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THE

DESIGN AND CONSTRUCTION

OF

CAMS

BY

ollansbee
CHARLES F. SMITH, FREDERICK A. HALSEY
AND OTHERS

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PREFACE

THE contents of this book, up to but not including Chapter X, describe the methods used by Mr. C. F. Smith, and the illustrations up to the same point are from drawings by him. The analysis and description of these methods are by Mr. F. A. Halsey, as is the section on the monotype cams. Other chapters are credited to their authors in foot-notes.

Mr. Smith has been connected with the construction of machinery involving the use of cams for twenty-five years, during fifteen of which he has made the design of such machinery a specialty. As a result of this extended experience he has developed the line of procedure which is herein explained and which is in the highest degree successful. He has designed machines containing as many as twenty cams and his experience is uniformly that given in connection with the chain-making machine illustrated in later pages. The cams are laid out on paper and constructed and key-seated from the drawings, and when assembled they perform their work in entire harmony and without filing, doctoring, or changing.

Such results need no comment and it is particularly fortunate that, instead of basing the explanations on hypothetical cases, it has been possible to use as illustrations the cams of a highly successful machine.

Of course, no single machine exhausts all the varieties of cams and, after showing how the cams of the chain machine were laid out and made, other varieties of cams which are not included in it are shown. Up to Chapter X these are all drawn from Mr. Smith's fund of experience and hence in no sense represent hypothetical conditions.

The prejudice that exists in many quarters against the employment of cams is unfounded. The movements which they are capable of giving is unlimited, and in cases in which other means will give the movement it is often at the expense of increased complication and trappy construction. Properly designed on the lines laid down, nothing can be more satisfactory than the operation of cams, the design of which is not a matter of good luck or of expensive experiment, but results directly from the intelligent application of general principles.

The descriptions and illustrations explain the entire process of cam design and construction, beginning with the preliminary charting and following through the determination of the forms of the cam grooves and their transfer from the drawings to the templets, the milling of the grooves, and the cutting of the key-seats. They thus, it is believed, fill, and fill well, a gap in machine shop literature which has long waited to be filled.

The later chapters point out methods which differ in some respects from those used by Mr. Smith, and hence add to the completeness of the presentation of the subject.

176689

CONTENTS

CHAPTER I

PAGE

CLASSIFICATION OF CAMS AND ORDER OF WORK 1
  Classification of Cams 1
  Order of Work when Laying Out Automatic Machines 1
  Order of Work when Laying Out and Making Cams 2
  Positive Motion *vs.* Spring Return Cams 3

CHAPTER II

THE MACHINE FROM WHICH THE ILLUSTRATIONS ARE DRAWN 4
  Laying Out a Single Cam 4
  The Limiting Angle 6
  Periphery Cams with Radius Arms 7
  Objectionable Forms of Cams 7
  Face Cams 7
  Two-Step Cams 8

CHAPTER III

THE OPERATION OF THE CHAIN-MAKING MACHINE 12

CHAPTER IV

CHARTING THE MOVEMENTS 16

CHAPTER V

LAYING OUT AN ACTUAL FACE CAM, MAKING THE FORMER AND MILLING THE CAM 19
  Locating the Zeros of the Individual Cams 20
  Transferring the Cam Groove Profile to the Templet 20
  Making the Former 22
  Key-seating the Shafts 22
  Key-seating the Cam Blanks and Locating their Zeros 24
  Milling the Cam 24
  Retrospect 24

CHAPTER VI

LAYING OUT AND MAKING PERIPHERY CAMS 26
  Locating the Zero of the Former 28
  Radius Milling Attachment 28

CHAPTER VII

PAGE
RAISED PATHWAY, YOKE, AND CONICAL CAMS 31
Raised Pathway Cams 31
Yoke Cams 31
Conical Cams 31

CHAPTER VIII

CAMS FOR PRESCRIBED MOVEMENTS 37
A Combination Movement 38
Determination of the Path of Movement by Cams 40

CHAPTER IX

REPEATED AND RETURN MOVEMENT CAMS 44

CHAPTER X

CHARTS WITH A SEPARATE BASE LINE FOR EACH CAM, ABBREVIATED CHARTS, EXTREME ANGLES, LOCATING KEY WAYS 46

CHAPTER XI

THE DOUBLE-CAM SYSTEM OF THE MONOTYPE 51
Modification of Cam Curves for High Speed 51
Laying Out the Cam Curve from the Chart 53
Outlining the Master Former 54
The Cam-Cutting Machine 55

CHAPTER XII

CAM MOVEMENTS OBTAINED FROM BASE CURVES OTHER THAN THE CIRCLE 58
Circular Arcs and the Ellipse as Base Curves for Cam Profiles 58
The Gravity Curve as a Base for Cam Profiles 61

CHAPTER XIII

THE LOCATION OF LEVER FULCRUMS FOR FACE CAMS 65

CHAPTER XIV

MINUTE ADJUSTMENT OF CAM LEVER MOVEMENTS 67

CHAPTER XV

GRINDING CAMS 69

THE DESIGN AND CONSTRUCTION OF CAMS.

CHAPTER I

CLASSIFICATION OF CAMS AND ORDER OF WORK

In any discussion of cams we must distinguish two kinds. The first is that in which the only requirement is that the operating point shall move a given distance, the movement beginning and ending at certain points in the revolution of the cam shaft. The second is that in which there is an additional requirement that the operating point shall at all times during its movement occupy predetermined positions. The first kind is much the more numerous and its discussion will form the chief subject of what follows, though the methods of laying out the second kind will also be shown. It will be noted that when laying out the first kind we are at full liberty to lay out the cam groove with a view to avoiding abruptness of movement, whereas, with the second kind, the movements being prescribed, we have much less liberty in this respect. It might be imagined at first thought that we have no such liberty, but this is not so, since with this, as with the first variety, more favorable angles of the cam grooves may be obtained by increasing the diameters of the cams.

Order of Work when Laying Out Automatic Machines

The starting point when designing any piece of automatic machinery is the work to be done. The piece to be operated upon is to be put through certain processes involving movements of the piece itself and of the chucking devices and other tools which operate upon it, and the sequence of these movements must first be worked out. The procedure to be described is to first design the end of the machine which operates upon the piece to be manipulated, deciding upon the operations to be performed and the movements to be made, and from this to work backward to the cams which are to give these movements and then to build the frame around the mechanism so designed. Of course, in this kind of work, like all others, the element of cut and try enters, and developments during the later stages of the process often make it necessary to retrace one's steps, but, broadly speaking, the order of work is as described:

1. Determine the sequence and character of the movements at the point where the work is to be done.
2. Determine the locations and forms of the cams necessary to give those movements.
3. Fill in the frame around this mechanism.

ORDER OF WORK FOR LAYING OUT AND MAKING CAMS

In the matter of cam layout and construction, many years' experience has resulted in a regular line of procedure which follows the determination of the movements given in (1) above, and is as follows:

1. The movements are charted full size upon a single sheet, the base line of which represents the development of the entire circumference of the cam surfaces. Verticals are drawn from the base line dividing the circumference into parts of equal width, the divisions usually representing five degrees, though their number may be increased in the case of very particular work. The left-hand vertical is divided into parts of an inch and on these vertical lines the movements are laid out, a line, which is parallel to the base line except where the movements occur, being drawn for each cam and its length being such as to represent the entire circumference. The rising and falling of each line with respect to the base line represents the movement of its cam roller in the cam groove. In laying out this chart we have given the already decided upon movements which locate the beginning and ending of the roller movements, and by connecting these points by inclined lines we get a picture of all the movements. These inclined lines will lie at various angles to the base line and at this point the first determination of the cam dimensions is made. To avoid undue abruptness of movement this angle should not exceed thirty degrees.[1] Should the steepest angle exceed this figure it is necessary to redraw the chart with such an increased circumference as will reduce the angle to this figure.

The diameter of the cam having this steepest angle being thus determined, this diameter is then preferably made the diameter of all the others, the result being that thirty degrees becomes the most unfavorable angle of the set of cams, all but one having, usually, a less angle than this. Making the cams of a single diameter also greatly facilitates their laying out in detail.

2. The movements being charted, the next step is the drawing of the cams and the laying out of their grooves, the grooves being so shaped as to give an easy start and stop to the piece, and for this a uniform method is used which gives a movement equivalent to that produced upon a cross head by the half turn of a crank having a Scotch yoke connection, and this movement is always secured at the operating point regardless of the disturbing influence of connecting rods, bell cranks, etc.[2]

3. The cam profile as thus determined is transferred to a piece of thin sheet zinc tacked over the drawing and then removed and carefully dressed to the required form by hand, thus forming a templet of the outline of the cam.

4. The templet is placed upon the blank which is to become the former for the milling machine and is scribed around with a fine, sharp scriber, thus transferring the outline to the former.

5. The former is then dressed to the outline thus obtained, and

6. The former and cam blank are placed in a suitable milling machine and the cam groove is milled.

---

[1] As will be seen in later chapters some constructors materially exceed this figure.

[2] Later chapters show methods which differ from this.

Positive Motion vs. Spring Return Cams

All of these steps will be fully described and illustrated in what follows. One important point in all cam work is to have the cam shaft of ample size to do its work without vibration, as thereby the life of the cams is greatly increased. The use of springs for the return movement should be avoided, the cam being made to perform this movement positively. In some classes of work, notably paper handling in which the material must be coaxed rather than driven, the use of springs is sometimes necessary to perform the *operating* movement, the return being effected by the cam, but in all ordinary metal-working machinery the cam may be made to perform both movements. Not only does this result in positive and correct movements regardless of speed, but it relieves the cams and rollers of unnecessary labor. When springs are used they must be strong enough to effect the movement with some surplus, and the effort of extending or compressing the springs is added to the effort necessary to do the work, thus in many cases more than doubling the pressure on the rollers. A notable instance of the use of springs is found in the linotype, in which the springs perform the actuating movements, the cams merely returning the parts to the idle positions. Of course, when springs are used the form of the cam surface down which the roller runs while the spring is doing its work has to be determined with the same care as when the movement is positive, the difference being that with the spring the roller merely rides on the cam under spring pressure, while without it the roller is embraced by the cam groove. Except that the milling cutter must be kept accurately to size, there is no greater difficulty in making a cam to give a positive return than in making one to be used with a spring.

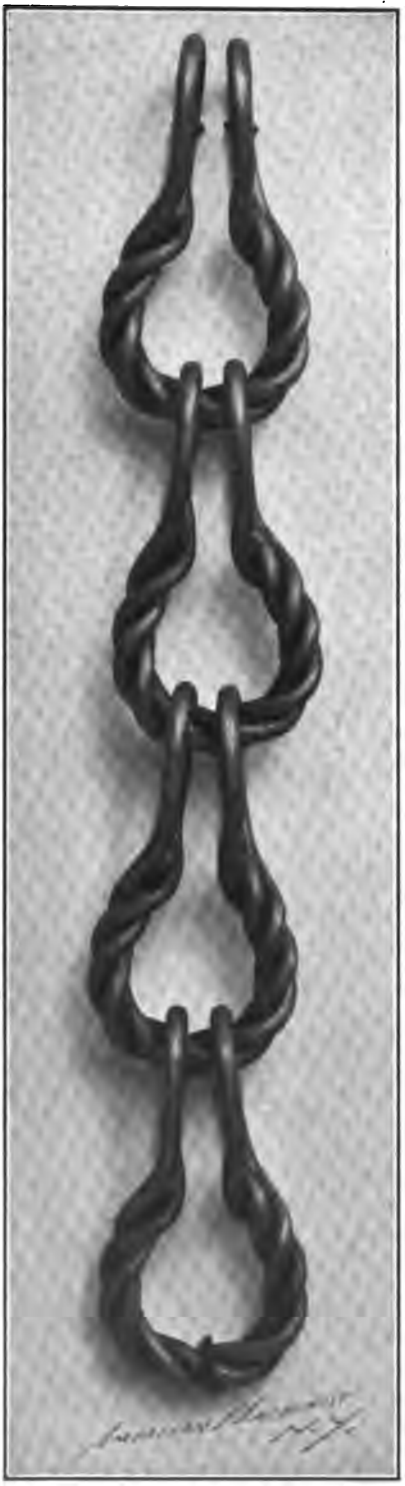

FIG. 1 The Chain to be Made.

CHAPTER II

THE MACHINE FROM WHICH THE ILLUSTRATIONS ARE DRAWN. LAYING OUT A SINGLE CAM

Figure 1 illustrates, full size, a new chain which was invented and patented by another party and then brought to Mr. Smith with a commission to design a machine for making it. The resulting machine, made by the Garvin Machine Company, is shown in Figs. 2 and 3. It contains sixteen cams and the sufficiency of the methods to be given is shown by the fact that the cams were laid out and constructed, including key-seating, by these methods, and when assembled they performed their work perfectly. Other parts of the machine required slight doctoring, as is expected in such work, but, except for dressing off the burrs, there was not given to the cams so much as the touch of a file after they left the milling machines. While the design of the machine as a whole cannot be given, some of its cams will be taken as examples of the methods used.

Laying out a Single Cam

While the charting of the movements is the first step in laying out a set of cams, it is desirable, in order to avoid a break in the description of the process of laying out the drawings and making the templets and formers, to here first take up the method of laying out a single cam groove without regard to its mates, in order to give an easy start and stop to the movements.

Figure 4 shows the simplest possible case, that of a periphery cam similar to those shown in the immediate foreground of Fig. 2, except that the roller is supposed to be mounted on a slide and thus to move in a straight line parallel with the shaft, instead of being mounted on an arm and thus move in the arc of a circle, as in Fig. 2. The rectangle 0, *B*, *C*, 12 represents the

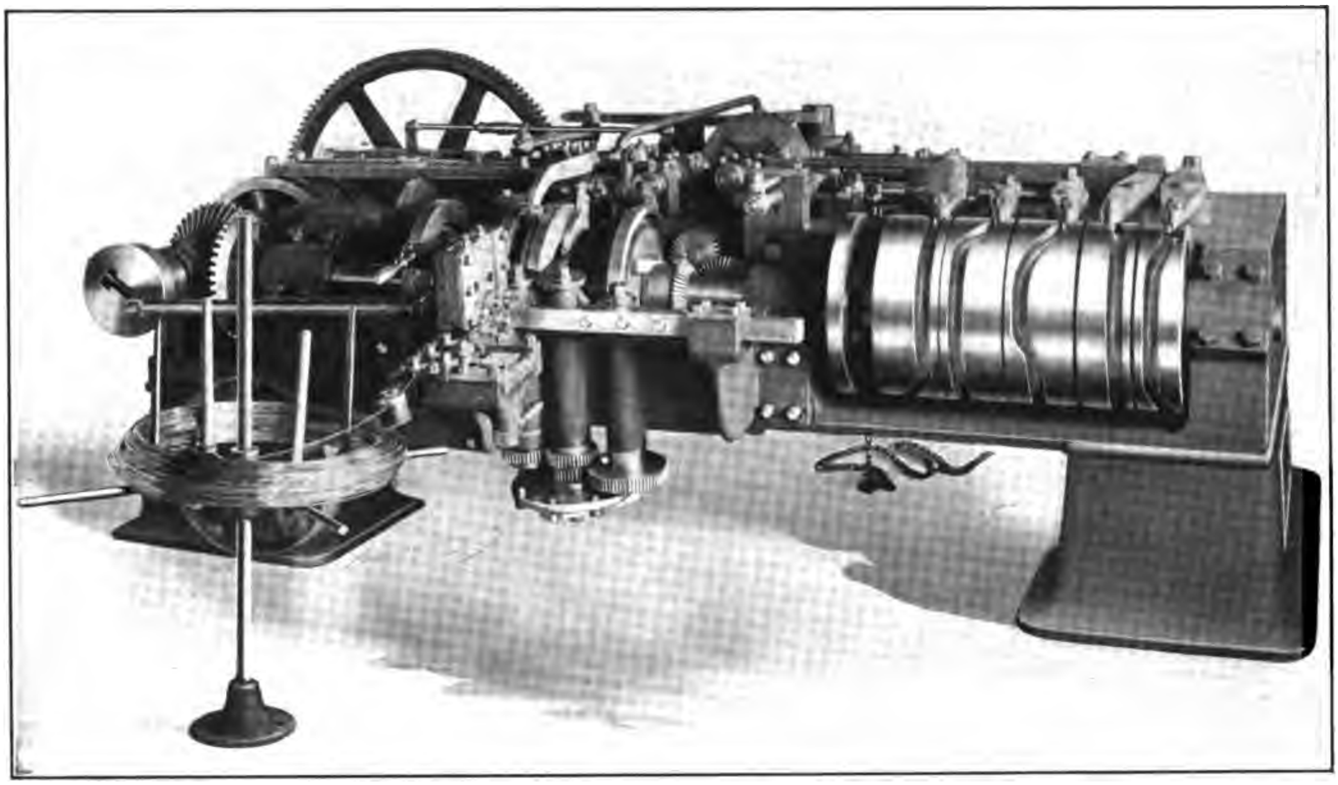

FIG. 2 Chain Making Machine. Front View.

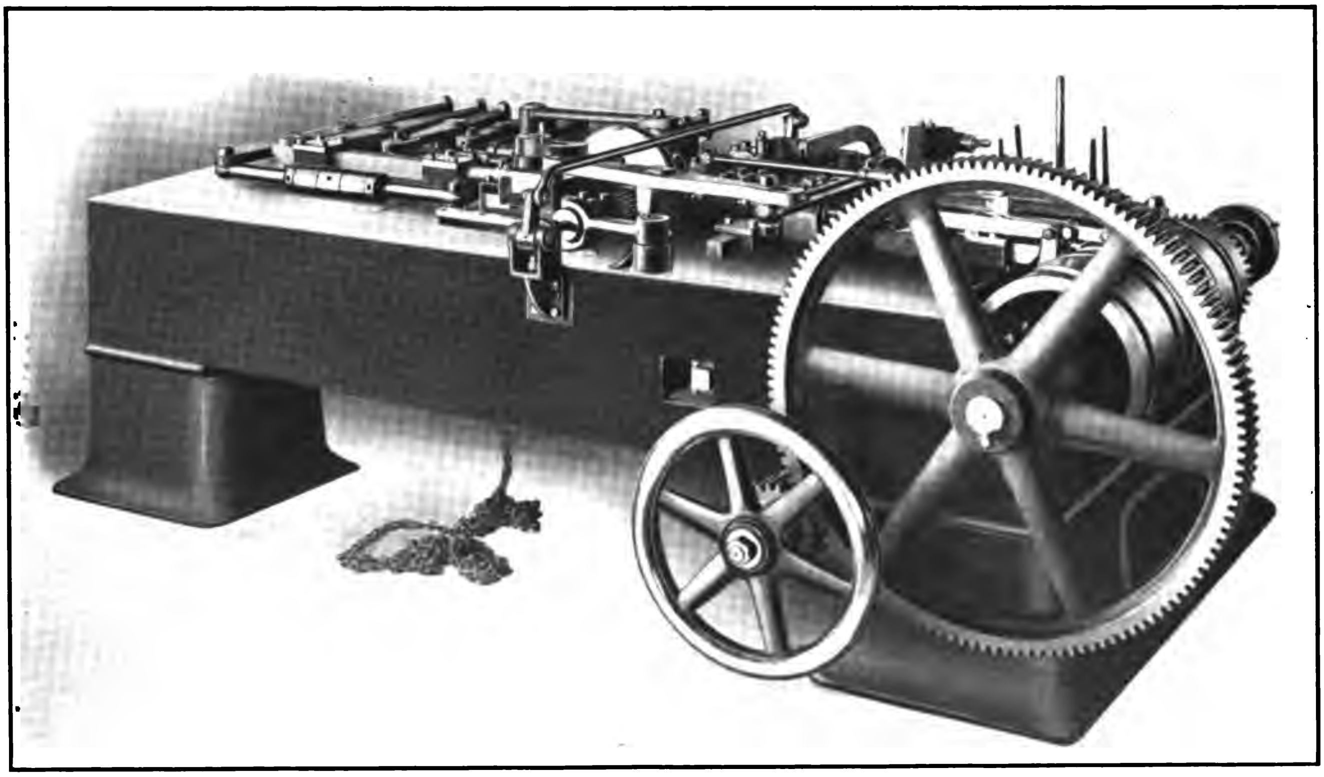

FIG. 3 Chain Making Machine. Rear View.

development of that part of the cam's periphery in which the movement is to take place. The movement of the roller is shown full size by the line o *B*, and this movement is to be performed while the cam turns through an arc of which *B C* is the development. Upon *A* o, equal and parallel to *B* o, a semicircle, called the throw circle, is drawn equal in diameter to the movement of the roller, and this semicircle is divided into any number of equal parts — in this case 12. The developed arc *B C* is then divided into the same number of equal parts and projection lines from the points on the semicircle give by their intersections with the corresponding verticals points on the center line of the cam groove as indicated by the small circles. In the present case the resulting curve is the projection of a true helix. It will be observed that while the cam turns from *B* to *C* the movement of the roller is precisely the same as would be given by a crank turning through the semicircle o, 6, 12. The return movement may or may not be performed in the same interval of time, but its groove will be laid out in the same way. Should the movements be performed in the same time the movement of the roller is precisely the same as that of a cross head, but with a pause at each end of the stroke, and should they be performed in different times the same will be true, except that one movement will be made more quickly than the other.

THE LIMITING ANGLE

The limiting angle of 30 degrees is that of the tangent to the middle of the center line of the groove as indicated in the figure. In laying out the drawing the distances o *B* and *B C* are given as has been stated, and it is desirable in the preliminary work to avoid the necessity of laying out the curve in order to determine this angle, as this might involve tedious repetition, and this determination may be made in several ways. Thus with the angle equal to 30 degrees there is a constant ratio of 2.72 between the length of the lines *B* o and *B C*. B o being given by the conditions of the problem, it is only necessary to multiply it by 2.72 in order to obtain the length which *B C* must have in order to make the final angle 30 degrees. Of course, the length of *B C* considered as a fraction of the entire circumference is also known and from this the entire circumference and the diameter of the cam may be quickly determined. Thus, *B C* being 2.72 times the throw and occupying say *n* degrees of the circumference, we have:

$$\text{Circumference} = 2.72 \times \text{throw} \times \frac{360}{n}$$

$$\text{or diameter} = \frac{2.72 \times \text{throw} \times 360}{3.1416 \times n}$$

$$= \frac{312 \times \text{throw}}{n}, \text{ approximately.}$$

That is, the greatest throw of the set of cams in inches multiplied by 312 and divided by the number of degrees of the cam's circumference occupied by the movement gives the diameter of the cam in inches. If a protractor is at hand the length of *B C* may also be determined by the fact that with the angle of the tangent equal to 30 degrees the angle *C* o 12 will be equal to 20 degrees 12 minutes, or, for practical purposes, 20 degrees. This ratio of 2.72, and this angle of 20 degrees, are strictly correct for periphery cams, of which the rollers move in straight lines only, but they may be used for cams of which the rollers are guided by radius arms without appreciable error. For face cams these constants may be modified, as will be explained later. It will be observed also that near the point of tangency the curve and tangent coincide very closely and we may take advantage of this to obtain a graphical method which is accurate enough and which is constantly used by Mr. Smith. Having divided the end circle, lay out the triangle *a b c*, the height being projected from the stroke circle and the hypothenuse being drawn at 30 degrees, when the base gives the length of one of the divisions of the base line. The same result may, of course, be obtained from the triangle *c d e*. The chief use of the constants is in the preliminary layout of the chart, as will be explained later. When laying out the curves on the individual cam drawings the graphical method last described is recommended.

PERIPHERY CAMS WITH RADIUS ARMS

If the roller is supported, as it usually is, by a radius arm the procedure is slightly modified as shown in Fig. 5. The base line and throw circle are divided as before, but instead of drawing perpendiculars from the base line divisions, arcs of circles are struck through them with a radius equal to that of the proposed radius arm, as indicated by the line *a b* and centers *c d e*, etc., and the intersections of these arcs with the projection lines from the divisions of the throw circle form the locating points for the center line of the cam groove as indicated by the small circles. The angle of this line at its middle should, as in the former case, be not greater than 30 degrees, as shown, and is predetermined as in Fig. 4.

OBJECTIONABLE FORMS OF CAMS

Figures 6 and 7 show very objectionable forms of cam grooves, Fig. 6 being for a roller mounted on a slide, and Fig. 7 for one carried by a radius arm. There is here no gradual starting and stopping, the motion being uniform throughout and of course started and stopped abruptly, and therefore with a shock. Similar forms are often seen, though usually with rounded curves drawn in by the eye or from arbitrary centers at the beginning and ending of the incline. The ease with which a crank motion curve may be laid out should lead to its regular use.

FACE CAMS

The two diagrams shown cover practically all cases of cams of which the grooves are upon the periphery. When the groove is upon the side or face of the cam disk the procedure is shown in Figs. 8 and 9. Fig. 8 corresponds with Fig. 4 in that the roller is mounted upon a slide which moves radially with the cam disk. The cam is here required to move the roller the distance o *B* while turning through the angle *B C*. The throw circle o, 6, 12 is drawn with a diameter equal to the radial movement of the roller and is divided as before into equal parts. The cam angle is similarly divided and radius lines o, 1, 2, 3, etc., are drawn. The division points of the circle are projected to its diameter, and arcs struck from the center of the disk and from the feet of the projection lines give by their intersection with the radial lines the points of the center line of the cam groove; the limiting angle appearing as before. In determining this angle for this style of cam the lower triangle *a b c* should be used in preference to the upper one *c d e*, as the tangency is closer below the middle point than above it. As with the periphery cam this angle is regulated by varying the diameter of the cam disk.

In laying down this style of cam upon the chart, the figure *B*, 0, 12, *C* must, of course, be represented by a rectangle of which the length is properly made equal to the outer arc *B C*. The constants which have been given for periphery cams become for this construction 3.23 and 17 degrees 15 minutes, or, for practical purposes, 17 degrees; that is, the length of the rectangle should be at least 3.23 times its height, and the angle of its diagonal with its base should not exceed 17 degrees, showing that face cams must be larger than the periphery style in order to have an equally favorable angle, and hence the latter are to be preferred.

While the constants given above apply to face cams, the extreme radius of a face cam groove is not much less than the radius of the piece, there being but a wall of metal and the radius of the roller between, and the convenience of a uniform outside diameter of cams is so great that it may be neglected, the curves of a chart being laid out as though the cams were all to be of the periphery style, both face and periphery cams being made of the same outside diameter. The face cam grooves will thus be slightly steeper than if they were of the periphery style, but in only a few, and probably in no, cases will the angle exceed 30 degrees. Were this angle to be materially exceeded in one of the face cams, especially in an important one doing heavy work, the whole set should be redesigned and enlarged.

In Fig. 9 the roller is carried by a radius arm of a length *a b*. An arc *c d* is so struck as to pass through the center of the cam shaft if possible.[1] Sometimes this is impossible, but the practise should be departed from only in case of necessity. The center *a* located, the arc *e f* is struck from the center of the cam disk, and from centers located on it and with a radius equal to the length of the arm the arcs *B* 0 and *C* 12 are struck, such that the arc *i j* is the angle of cam movement during which the roller movement is to take place. The arc *e f* between the extreme centers *g h* of the arcs B 0 and *C* 12 is then divided as in previous cases, and arcs 0, 1, 2, 3, etc., are drawn from the division points as centers. The throw circle 0, 6, 12 is then drawn and the cam curve is quickly located. The limiting angle is again shown.

Figures 10 and 11 illustrate face cams giving uniform motions, the former operating a roller mounted upon a slide and the latter upon a radius arm. As before said, these forms should be avoided, except of course when the motion required is such as to necessitate their use. They are sometimes used in combination with crank-motion forms, the crank motion being used to bring the parts to speed and also to bring them to a stop. An example of this will be given later.

TWO-STEP CAMS

Figure 12 illustrates the laying out of a face cam to give a movement in two steps, each step being taken with a true crank motion. The driven crank arm is centered at *A*, its extreme positions being *B* and *C* with an intermediate dwell at *D*. The first movement

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[1] Later chapters show methods which differ from this.

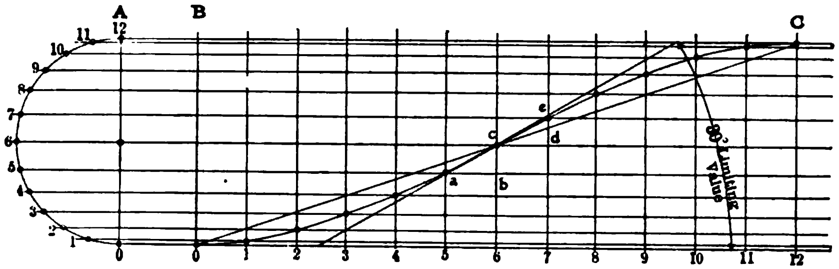

FIG. 4 Periphery Cam. Crank Motion, for Operating a Slide

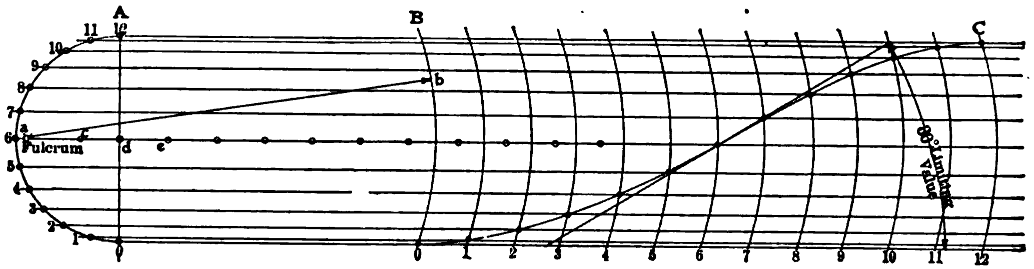

FIG. 5 Periphery Cam, Crank Motion, for Operating a Pivoted Lever

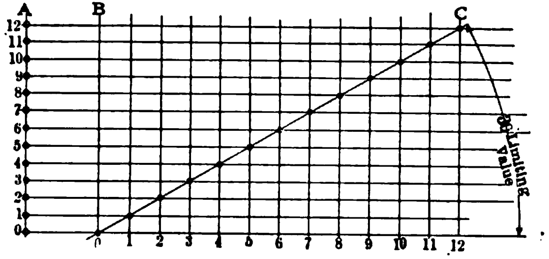

FIG. 6 Periphery Cam. Uniform Motion, for Operating a Slide

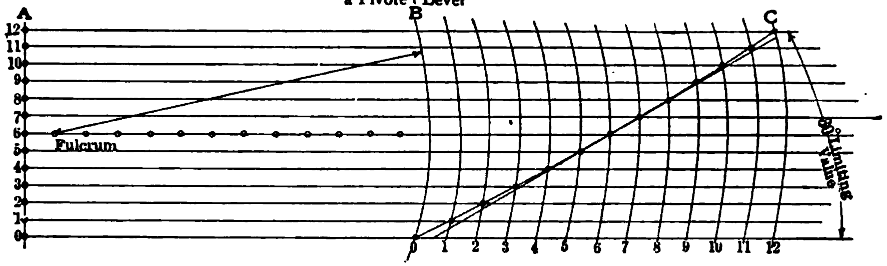

FIG. 7 Periphery Cam. Uniform Motion, for Operating a Pivoted Lever

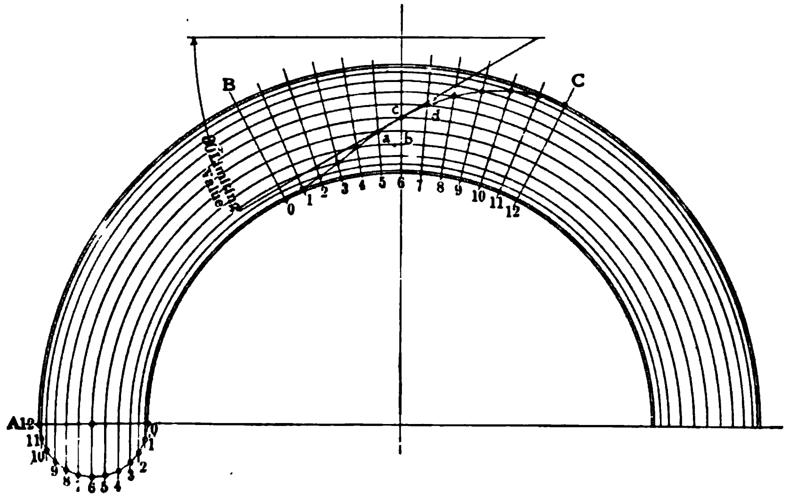

FIG. 8 Face Cam. Crank Motion, for Operating a Slide

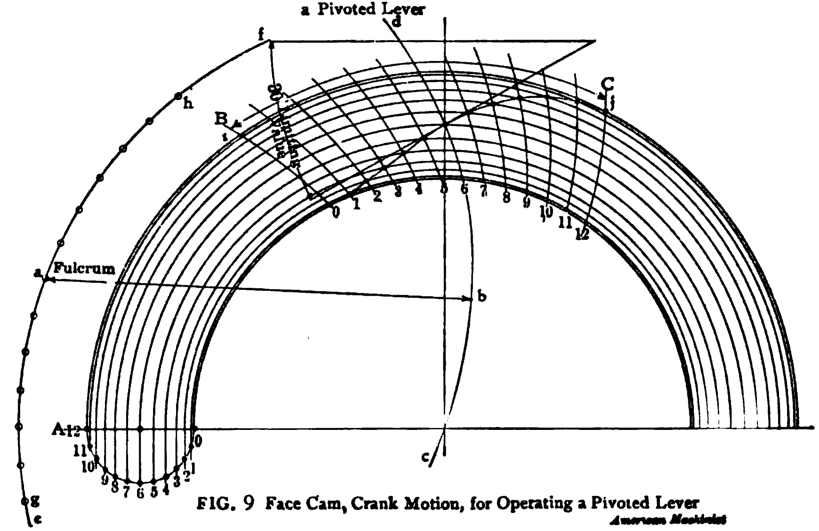

FIG. 9 Face Cam, Crank Motion, for Operating a Pivoted Lever

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THE LAYOUT OF CAM GROOVES

is from *B* to *D* and the second from *D* to *C*. The cam roller is carried by an arm pivoted to the end of the crank arm and having a forked end which straddles a square guide block which rides on the shaft. A construction of this kind is shown in Fig. 2 at the left of the bevel gears in the middle of the front side of the chain machine.

The extreme throw is laid out on the horizontal center line and the lines *B* and *C* are drawn from *A* to give the same movement on each side of the vertical center line. The arc *E F* is drawn and the position *D* of the arm at the intermediate dwell is laid down. Chords *G H* are drawn and on them throw circles are drawn each of which is divided into parts as before, the division points being projected to the arc *E F*. The extreme positions of the arm which carries the roller are drawn in at *I* and *J*. The extreme positions of the roller are at *K* and *L*, and an arc *K L* tangent to *J* will give approximately the path of the roller, which may thus be treated as though pivoted at the center *M* of the arc *K L*, which does not conform to the recommendation that it should pass through the center of the cam shaft, as it is impossible to make it so conform. With radius *F* o *K* and centers at the points of the arc *E F* projected from the division points of the throw circle on chord *G*, the positions of the circles 0, 1, 2, 3, etc., from the cam-shaft center are obtained and they are drawn. Points *P Q* are found on the arc *M N* from which to strike arcs 0 *a* and 12 *a* to include the angle of movement of the cam within which the first movement of the roller is to take place, and the arc *P Q* is then divided as before and arcs 1 *a*, 2 *a*, 3 *a*, etc., are drawn, the intersections of which with the correspondingly numbered circles drawn from the cam-shaft center give the outline of the center line of the cam for the first movement. The required length of dwell *R S*, for which the cam curve is of course a circle, is then laid out and the profile for the second movement is then determined in the same way from the throw circle drawn on chord *H*, but of this details need not be given. The inevitable limiting angle of 30 degrees appears in both cases.

The return movement, not shown, is laid out from a third throw circle of which the diameter is equal to the sum of those already used.

The above methods are sufficient to lay out any single-roller periphery or disk cam of the types illustrated of which the time and extent of the movement only are determined beforehand. The laying out of other special forms will be given later.

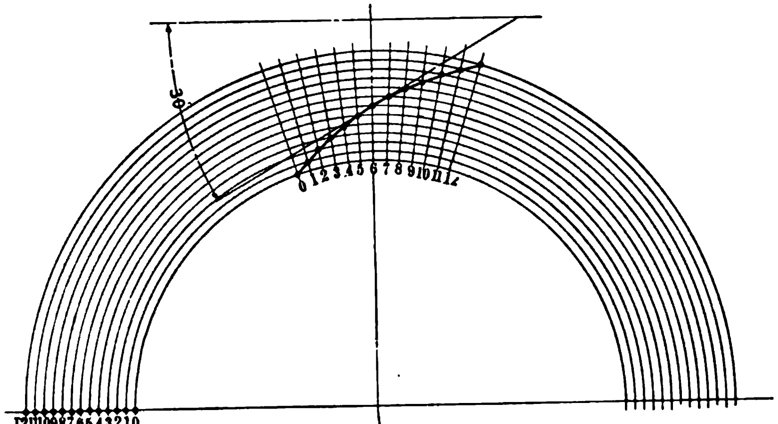

FIG. 10 Face Cam, Uniform Motion, for Operating a Slide

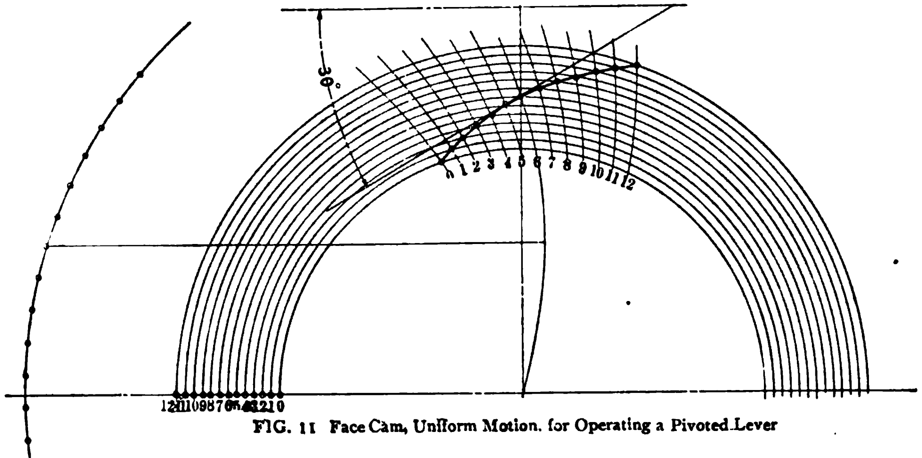

FIG. 11 Face Cam, Uniform Motion, for Operating a Pivoted Lever

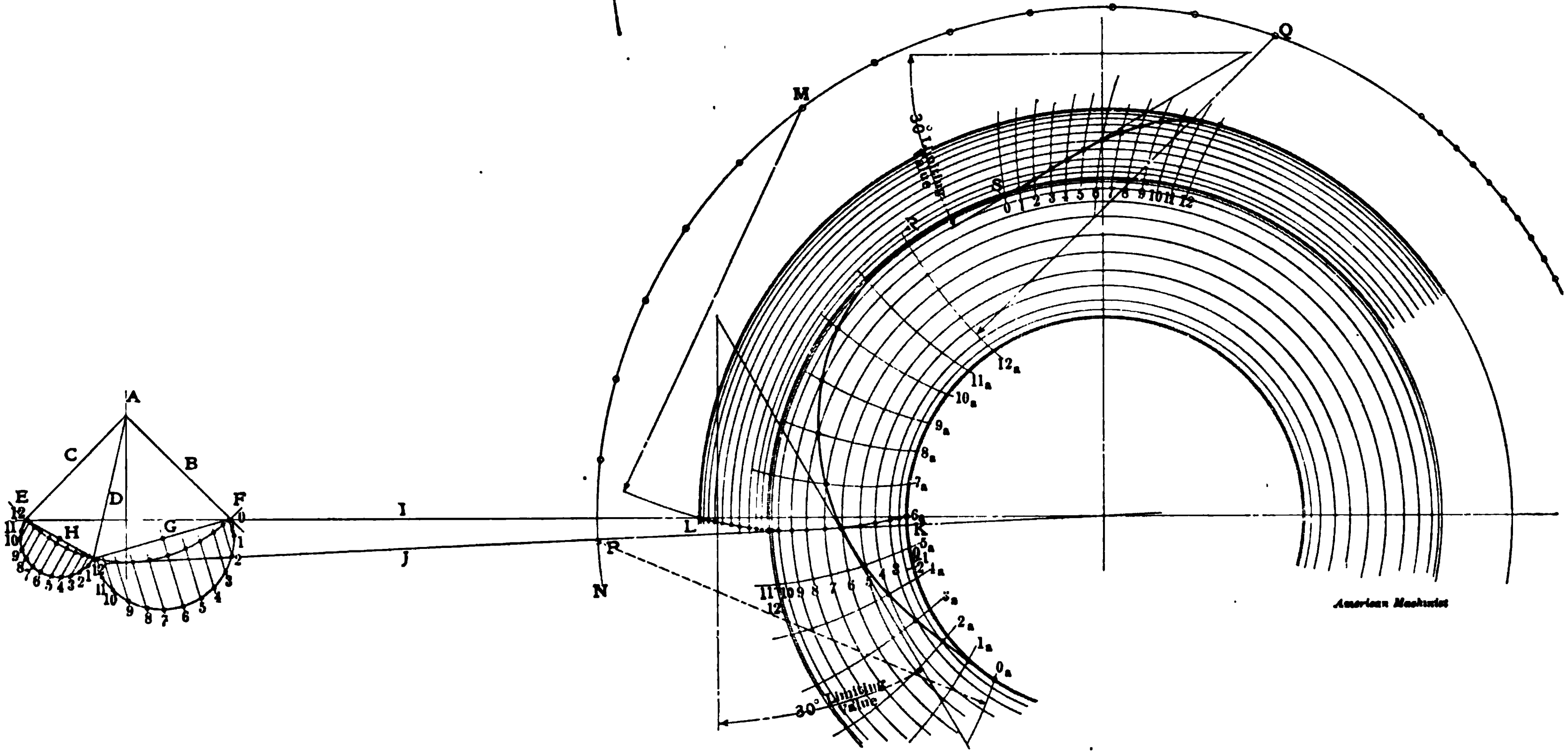

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FIG. 12 Face Cam, Crank Motion, for Double Throw Fork Lever

THE LAYOUT OF CAM GROOVES

CHAPTER III

THE OPERATION OF THE CHAIN-MAKING MACHINE

FIGURE 13 illustrates in its various stages of manufacture the chain of which a view of a complete section was given in Fig. 1. The first operation is to draw in and cut off the wire, the second to fold the ends, the third to twist the folded piece, the fourth to insert the twisted piece in the previous complete link, the fifth to bend the U, and the last to close the ends. While this completes the list of operations on the link, it does not include all of the movements. Thus, after twisting, the link must be turned through 90 degrees in order to insert it through the previously made complete link, it must be turned back again before bending, and it must be transferred bodily from the chucks in which the twisting was done to the position from which it may be inserted in the previously made link, in addition to which various gripping and releasing movements must be made.

The location of the wire reel from which the chain links are made will be seen in Fig. 2, together with the straightener rolls through which the wire passes. Within the machine is a grip to hold the wire in position after cutting off a link, and the first action when starting a fresh link is the releasing of this grip when a fresh length of wire is fed in. This feed of the wire is accomplished by the crank at the extreme left of Fig. 2, which operates a slide shown in the middle foreground. The slide has a rack cut upon it which drives a pinion as shown, which pinion by the gear and shaft connections, also shown, drives a ratchet and pawl below, whereby, during one movement, the feed rollers, which are seen just beyond the straightening rolls, are turned and the wire fed in, while during the return motion of the slide the feed rolls stand still. The ratchet is fitted with differential pawls which give sufficient accuracy.

A crank was adopted for the initial movement because, there being no movement behind it, there is ample time for the work to be done.

Figures 14 to 17 give bird's-eye views of the operations on the piece subsequent to cutting off. In the background of Fig. 14 the wire is seen passing at a slight angle through the bending chucks and gripped by a second grip, called plyers, between the chucks. Fig. 15 shows the same wire partially bent to the shape of the second illustration of Fig. 13. Fig. 16 shows it after being stripped from the plugs over which it was bent, and transferred to the twisting jaws which have partially twisted it. After twisting, the double-disk transfer chuck, which is seen edgewise in all the views, moves up to the twisted wire, grasps it in one of its four jaws, withdraws it, and in two steps turns half over and then moves toward the observer and presents the wire to the chuck seen in the immediate left foreground of Fig. 16. This second chuck then advances and grasps the twisted link, as also seen in Fig. 16, when the disk chuck withdraws, as seen in Fig. 17, the second

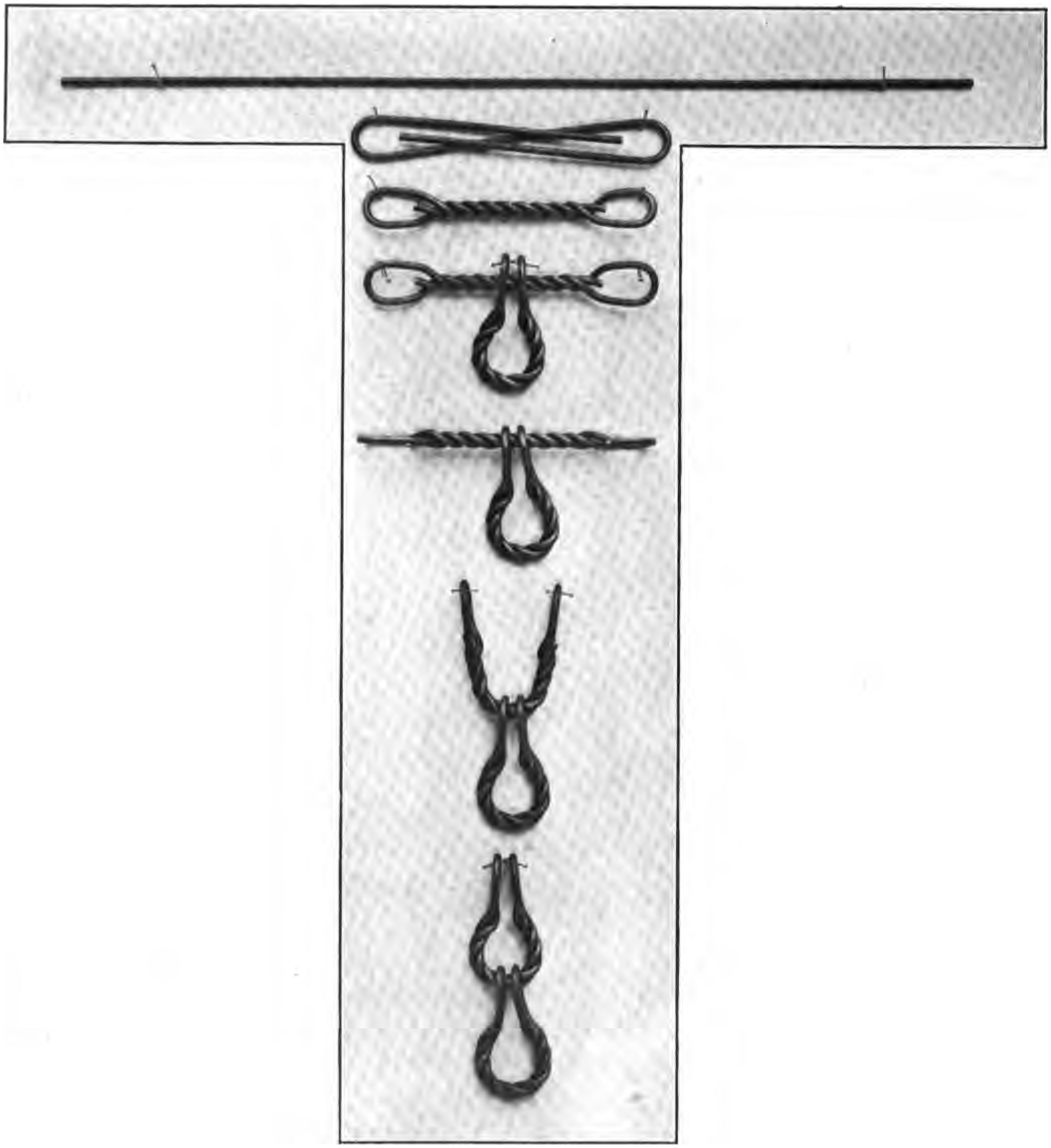


FIG. 13 Chain Links in Successive Stages

chuck turns the link through 90 degrees, as seen in Fig. 14, and advances it through the previously bent link, as seen in Fig. 15, when it turns the link back through 90 degrees and a third chuck at right angles to the second takes possession and the link is bent to the U form, as seen in Fig. 17, and, lastly, the link ends are closed — an operation which is not shown in the illustrations.

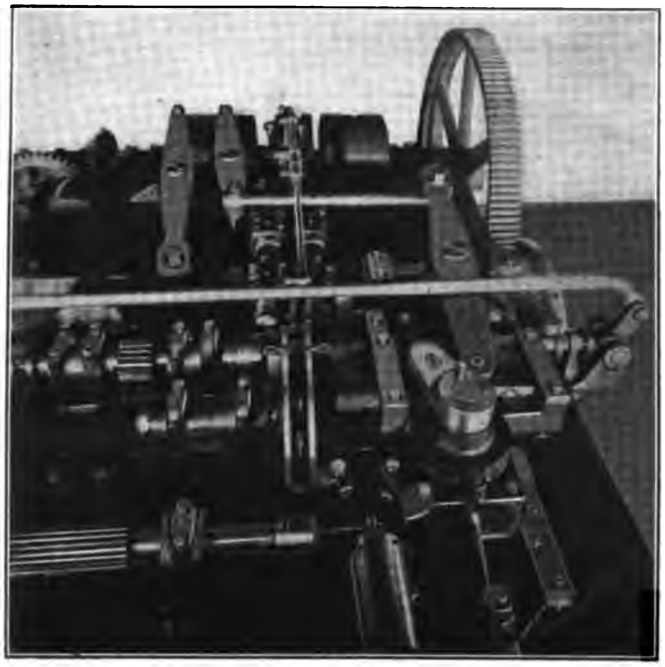


FIG. 14

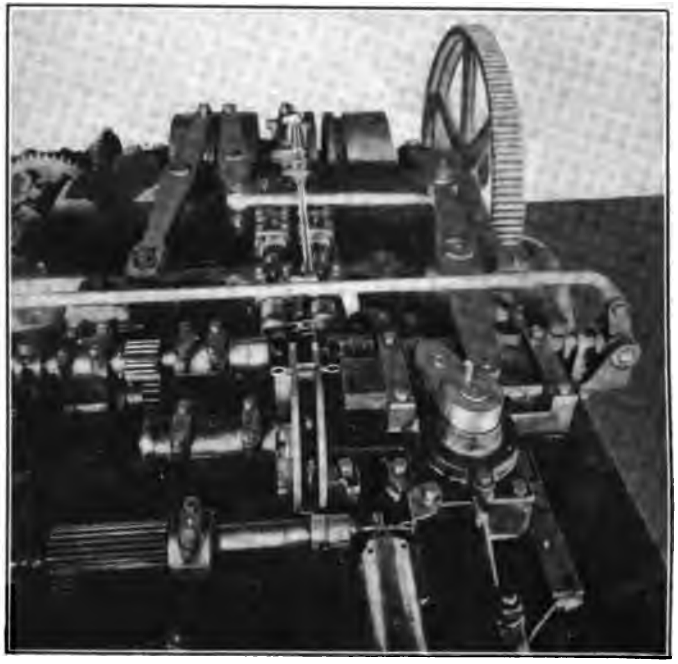
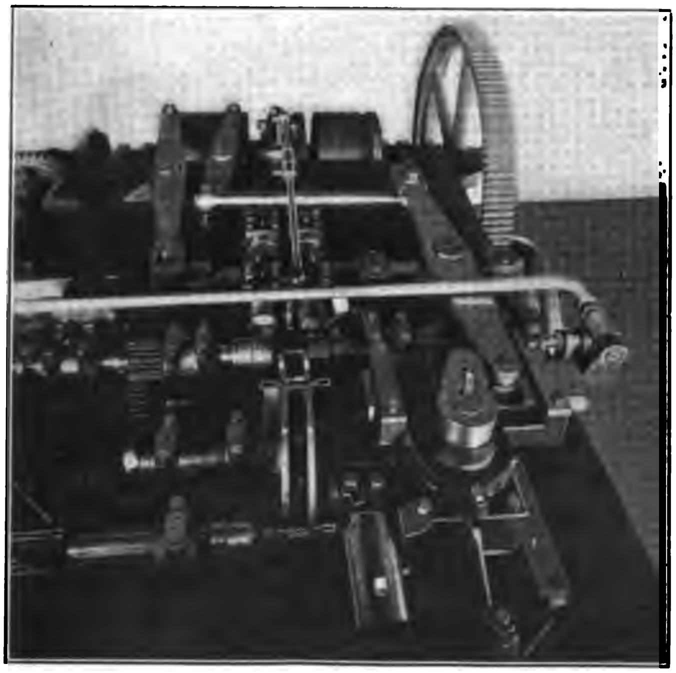


FIG. 15



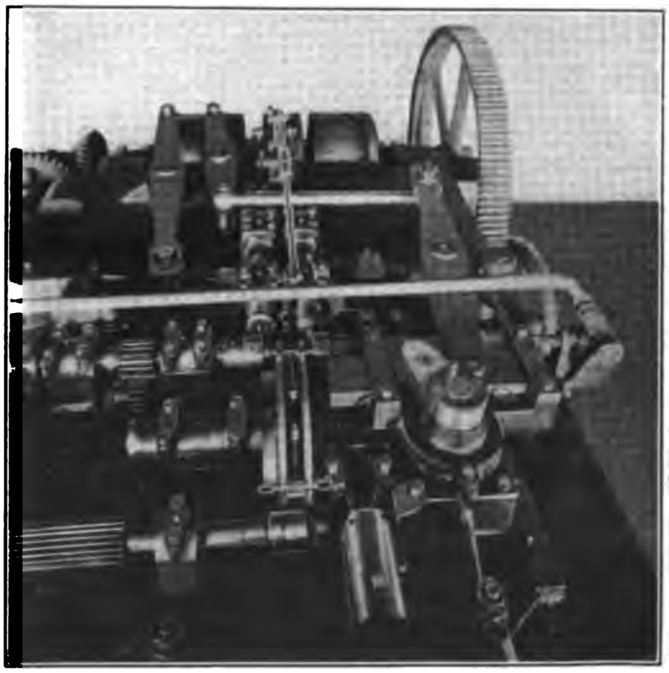


FIG. 16 THE OPERATION OF THE MACHINE FIG. 17

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in, and at all points of the curve the height above the base line gives the extent of the movement made at those points.

The starting point of this curve is made the zero line of the chart, the entire length of the chart representing the assumed circumference of the cams which must be assumed at the start, but which is, of course, subject to modification.

The assumption of a common zero line for all the cams is equivalent to the assumption that all the rollers will lie on a line parallel with the shaft, as those in the foreground of Fig. 2 in fact do. Other cams, however, have their rollers located on their sides or bottoms or at various angles, and such a disposition of a roller shifts the zero of its cam by the angle by which the roller is displaced from the assumed zero, and this must be taken care of when laying out the actual cam and locating its key-seat, as will be explained later. The assumption of a common zero when laying out the chart simplifies the work in large degree. The angles of displacement of the different cams are, of course, determined from the positions of the rollers given on the general design of the machine, and are later on noted on the chart by the various verticals marked *retainer*, *former*, *linking head*, etc. As a matter of fact, in the chain machine, but one cam has its zero to correspond with the chart.

After the wire is fed in, the next movement is to retighten the wire grip, which is indicated by the lowest cam line of the chart marked 8 and also by *wire grip on*, which shows by its location that the movement of the gripping cam is completed just as the movement of the wire into the machine is finished. Tracing this line to the right, we find it to be straight, indicating that the grip is held closed until near the end of the chart, when the dropping of the line at the note *wire grip off* shows that the grip is loosened just as the next movement of the wire into the machine begins.

Simultaneously with the tightening of this grip the second grip called *plyers* in the chart and shown between the bending chucks in Fig. 14 must be tightened in order to retain control of the cut-off piece, and this is done as indicated on the chart by the line *plyers hold wire*. At the completion of this movement the first operation on the wire — cutting it off — is made as shown by line 7 *cut off*. The height of this line above the base line, like that of all the others, shows the actual movement of the cam roller, in this case ⅞ inch, which is reduced by a lever to ⅛ inch at the nipper jaw.

The throws of all the cams are obtained in this way by working backward from the movement of the operating point through the levers and other connections to the rollers, and the laying out of the chart then consists in giving to each cam its required extent of movement and in making it begin and end this movement at the right time. The movements are individually simple, being the simple shifting of a lever, the precaution being always observed that the cams are of such size that the most abrupt movement does not involve a steeper angle than 30 degrees. The laying out of cams thus becomes a matter entirely separate from and subsequent to the design of the machine as a whole. The operating parts and their movements and the location of the cam shaft being determined and the connecting levers laid down, the matter, so far as it relates to individual cams, reduces itself to the moving of these levers at the right time and by the right amount. The chart deals with these movements only, without regard to their direction or the connecting mechanism.

The next operation is the bending of the link into the form shown in the second view, counting from the top of Fig. 13 and in the act of being done in Fig. 15, and this is done as shown in the chart by line 5 *winders wind link,* the word winder being used for this operation because bender is used for the binding of the link into U shape, as shown by line 15 *bending link,* the number being at the left and the words at the extreme right of the chart.

It would be tedious and unnecessary to trace through all of the motions in detail, but to avoid confusion it should be noted that in the present machine two revolutions are required to complete a link, and the chart thus represents two revolutions, the first revolution completing the bending of the link into the form shown by the second view of Fig. 13, and also getting the twister jaws into position for twisting. A length of wire is fed in and a link is completed at each revolution, the machine operating on two links at all times.

CHAPTER V

LAYING OUT AN ACTUAL FACE CAM, MAKING THE FORMER AND MILLING THE CAM

To illustrate the layout of an actual face cam, No. 3, *stripper moves full forward* will be shown, this being the cam which strips the link from the plugs over which the ends are bent. The cam and its relation with the machine frame are shown in Fig. 19. It is the third cam from the large gear wheel in Fig. 3. It moves a slide *a* by means of a fulcrumed lever *b*, the position of the roller center being at *c* on an arc which is laid out to pass through the center of the cam shaft, as already advised in connection with Fig. 9.

The diameter of the cam disk being that of which the length of the chart is the circumference, we draw the outer circle of the cam and lay down the thickness of the outer retaining wall, ¾ inch, and the radius of the roller, also ¾ inch, and draw the arc *n o*. The height of the cam curve above the base line of the chart, 1¾ inches, gives the radial movement of the roller, and we lay it down as shown and draw the arc *p q*, giving the inner limit of movement of the roller center. The throw circle is next drawn in and divided as has been described, and arcs from the feet of the perpendiculars from the division points are drawn. The arc through the center of the throw circle gives, by its intersection with the arc struck from the fulcrum *b* of the lever as a center, the location of the mid-position *c* of the roller. In the present case this happens also to be the middle of the return curve of the cam, but this is accidental.

Consulting the chart, we find that the roller of this cam must begin its movement 270 degrees after the beginning of the feeding inward of the wire, the foot of the cam curve lying on the line which is 270 degrees removed from the zero line. The chart also informs us that the working movement of the roller must be made during 40 degrees of movement of the cam. Measuring back, therefore, 270 and 40 degrees from the position *c* of the roller, as shown, we determine the positions of the arcs on which the beginning and ending of the cam curve must lie. Drawing an arc *d e* from the center of the cam shaft as a center, and passing through the center *b* of the lever fulcrum, we take its radius *b i* in the tram and find centers *f g* from which to strike arcs *j k* and *l m* passing through the cam shaft center and the 270- and 40-degree points. These arcs, by their intersections with *n o* and *p q*, determine the position of the cam curve for beginning and ending the movement. Dividing the arc *f g* as before, we find points 1, 2, 3, etc., from which to strike arcs 1, 2, 3, etc., which by their intersections with the arcs from the feet of the perpendiculars through the divisions of the throw circle give the successive positions of the roller center. From these centers circles having a radius equal to that of the roller define the cam groove. The return movement is laid out in the same way and from the same stroke circle, and is thus the same as the acting movement, although the two cam curves are very different. Those portions of the groove which represent dwells of the roller are of course arcs of circles having the center of the cam shaft as centers, and are drawn in.

Locating the Zeros of the Individual Cams

The zero of the chart should be looked upon as the zero of action and not, in general, of position of the cams; that is, the beginning of each cam curve is located the number of degrees behind the beginning of the feeding in of the wire that the movement of the cam must begin behind the movement of the wire. It is also the zero of position of the wire-feeding crank and were the rollers all located in the angular position of this crank at the beginning of its movement, it would also be the zero of position of the cams. Convenience, however, makes it necessary to locate the rollers in various positions around the shaft and thus to change the zeros of position of their cams.

The zero position of the wire-feeding crank is the outer center which is in the horizontal center line, and this point has been marked as zero in Fig. 19 — that is to say, the zero of Fig. 19 is the zero of the chart. The effect of measuring back from the position of the roller the number of degrees which the chart shows the cam to act behind the beginning of the movement of the wire, locates the cam on the drawing in the position which it must occupy when the movement of the wire begins. The angular distance from the zero to the position *c* of the roller is therefore the angle by which the zero of position of the cam differs from the zero of action as shown on the chart, and measuring it, we find it to be 202 degrees and we draw a vertical line *stripper* on the chart at this angle. This plan is followed with all the cams, and the horizontal center lines therefore locate the individual zeros from which the cams are located on the machine.

Transferring the Cam Groove Profile to the Templet

The form of the cam groove is now determined, and it remains to show how this form is transferred from drawing to templet, to former and to the cam blank, and how the location of the key-seat is determined, for it is perfectly easy to locate the key-seats in advance to the entire exclusion of set-screws.

The process of transferring the outline of the groove to the templet is shown in Fig. 20, which, however, does not show cam 3, of which the layout was given in Fig. 19, but cam 11, which, as shown by the chart, passes the twisted link through the previously formed complete link previous to bending. The selection of a different cam is perhaps of advantage, as it lends variety to the explanations.

A piece of thin sheet metal, for which zinc is preferable as it takes the lines readily, is tacked down on the drawing as shown by the shaded outline. This sheet is previously cut to a shape which shall fall within the pitch line of the cam groove but without the inner border of the groove. Its form is easily determined by laying a piece of tracing paper over the drawing and then drawing freehand a line which shall mark the desired outline of the zinc. The paper is then trimmed to this line and is used as a templet to which to cut the zinc.

With the zinc tacked over the drawing the horizontal and vertical center lines are carefully drawn with a fine, sharp scriber, and then with a pair of dividers having fine,

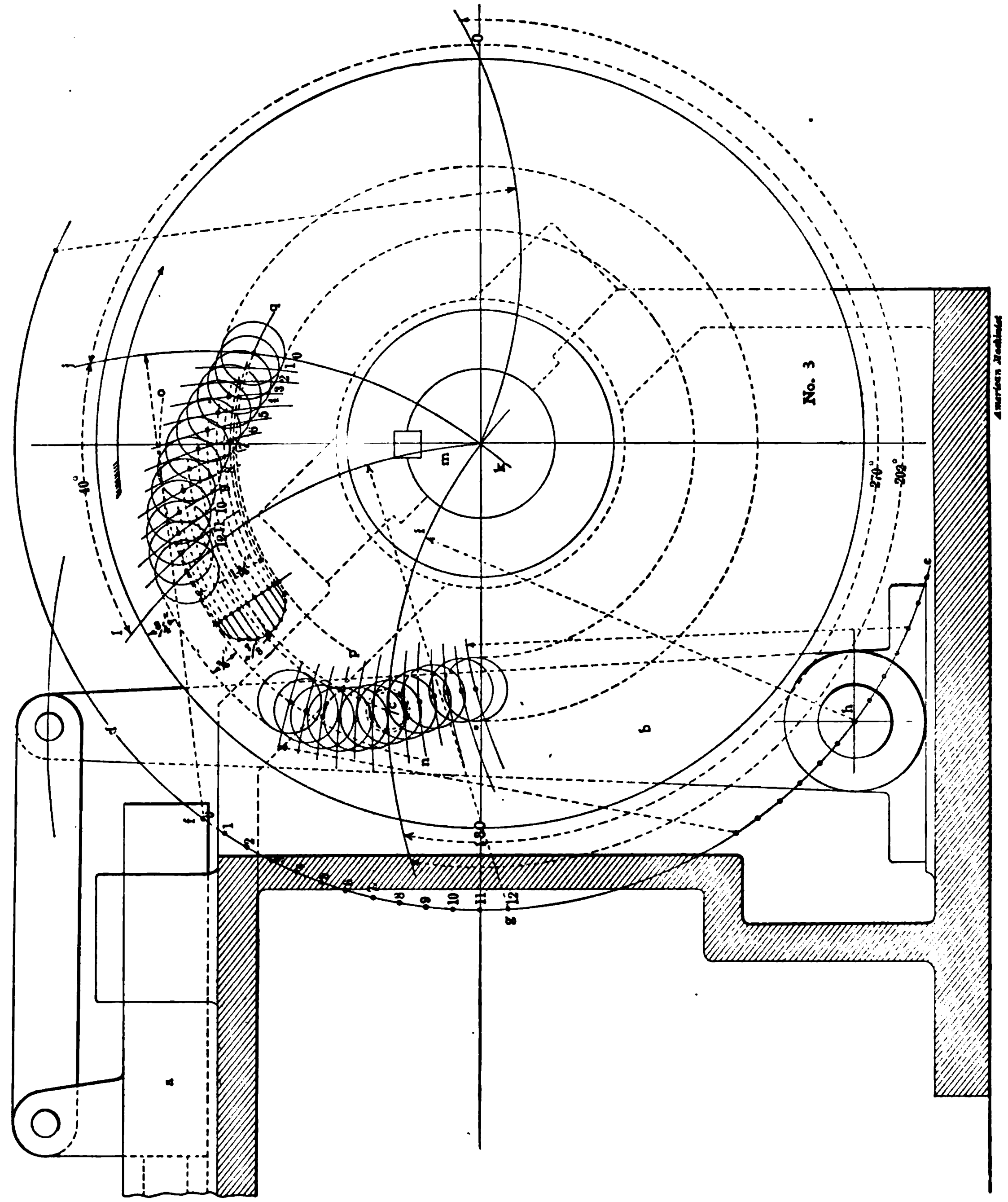

FIG. 19 Laying out a Face Cam

sharp points those portions of the roller circles which are covered up by the zinc are redrawn on the zinc. With an irregular curve the bounding line of these arcs is then drawn, though not shown in the illustration, the circular portions of the groove which represent dwells of the cam roller being drawn with the dividers from the center of the zinc as located by the center lines.

The extremities of the horizontal center line on the templet are marked 0 and 180 in order to properly locate the templet on the former blank after the zero has been located on the blank, as will be described later.

MAKING THE FORMER

The outline completed, the metal is carefully dressed down by hand to the outline and the templet is then placed upon the former blank which has previously been faced off. The center lines are drawn on the former blank and marked as on the drawing and templet and the outline is then transferred to the blank by a fine, sharp scriber, and the former is then dressed down to this line, the bulk of the metal being removed in a milling machine. The circular portions of the outline are easily followed exactly and the other portions are followed as closely as possible, after which they are dressed down to the outline as carefully as possible by hand.

The reader will observe that the cam curve is in this case drawn in dotted lines because of the fact that the roller, and hence the groove, are for this case on the far side of the cam. Were they on the near side the fact would be shown by drawing the curves in full lines. With the curve on the near side the templet would of course be laid on the former blank in the position in which it was laid out, whereas with the curve on the opposite side the templet must be turned the opposite side up. This is noted on the templet before it is removed from the board by the words *cut other side up*, as shown. The point of view when drawing all the cam curves should be from the same end of the shaft.

KEY-SEATING THE SHAFTS

Before doing any work in cutting the cam grooves the shafts are key-seated, as are also the cam blanks. In order to locate the key-seats in the shafts they are assembled in the machine with the connecting gears keyed to them, together with the wire-feeding crank, which is placed at the zero position. The mating teeth of the gears which connect the shafts are marked so that they may be reassembled at any future time in the same relation, and the key-seat is located by a surface gage and scriber. The bottom surface of the main bed, Fig. 19, is planed and parallel strips projecting out under and beyond the cam shaft are bolted to it. On these strips additional long parallels are laid, thus locating a base line which is truly parallel with the shafts. A surface gage supported on these strips is then used to locate the key-seat the entire length of both shafts. The actual location of this line around the shaft is unimportant, but for the sake of uniformity the exact top of the shaft, as indicated in Fig. 19, is recommended. The key-seats located, the shafts are removed and the seats are cut. To key the cams to the shafts, parallel keys made a sliding fit and secured from displacement endwise by set-screws bearing on the keys are recommended.

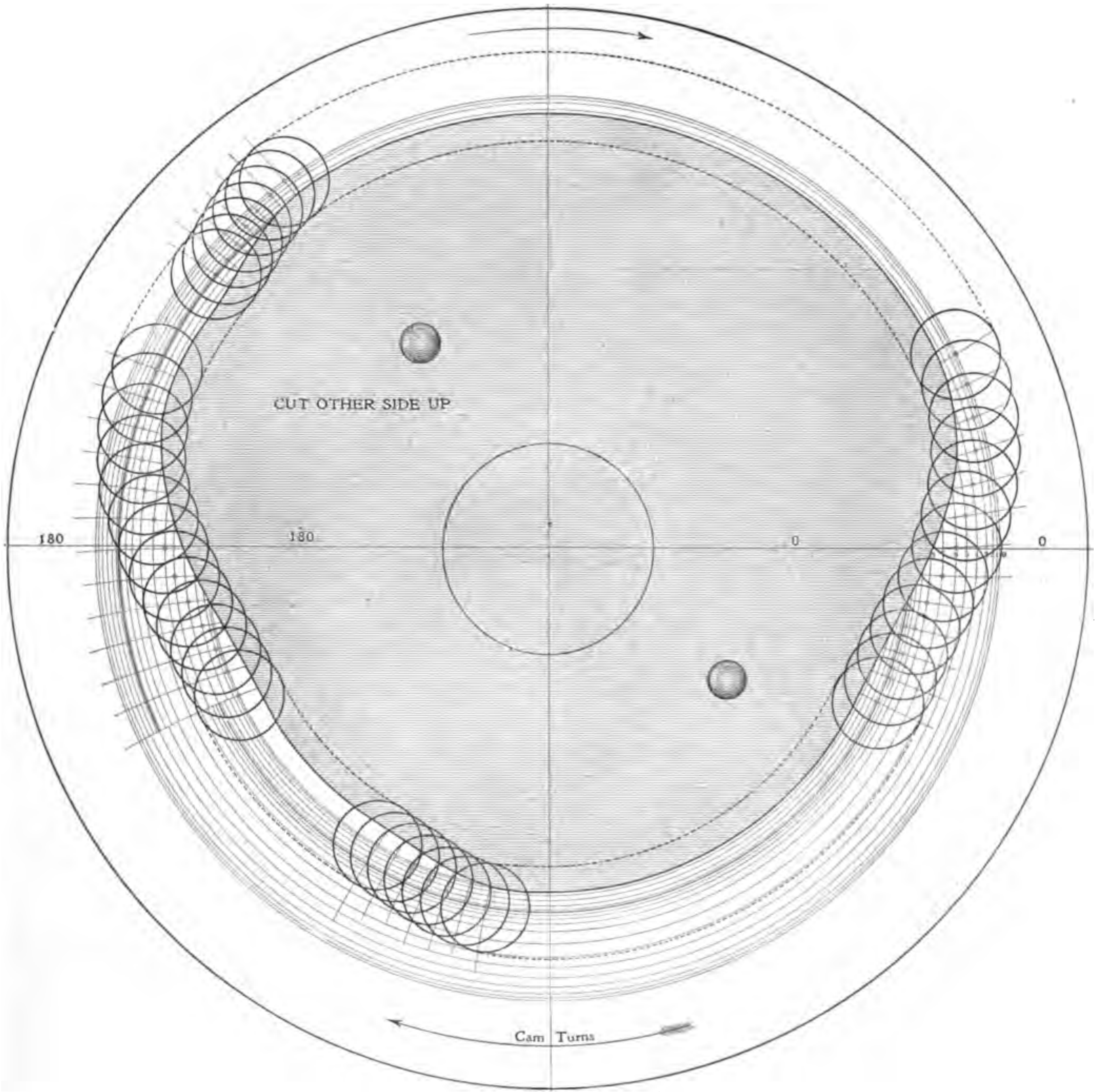

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FIG. 20 Transferring the Cam Profile to the Templet

Key-Seating the Cam Blanks and Locating their Zeros

The cam blanks are then key-seated, the location of the key-seats being of no importance, as the forms of the cams do not yet appear upon them. The blanks being numbered in accordance with the scheme of numbering shown on the general drawing and chart, they are placed on the shafts in their correct order, keyed thereto, and the shafts with the blanks are replaced in the machine preparatory to locating the zeros on the blanks.

To locate the zeros the parallels and surface gage already described are again used. The wire-feeding crank is located on its center and the surface gage is adjusted to the exact center of the shaft and then carried around the machine, scribing a line on each blank, which is obviously truly horizontal with the centers of the shafts and hence corresponds to the zeros of the chart and of the cam drawings, as shown in Fig. 19. These lines are scribed across the periphery of all blanks and also on the faces of the face-cam blanks, and are stamped zero in all cases.

Milling the Cam

Turning back to Fig. 19, it will be recalled that the corresponding horizontal zero line has been located on the drawing and transferred from there to the templet and to the former, and it only remains to place the former and cam back in the milling machine in the proper relation to insure a correct cam.

In the case of a face cam this is insured in the following manner: A cam blank and its former are placed on an arbor, and the zero lines are made to match, when a hole is drilled through both pieces. The face-plates of the cam-cutting machine have pins adjustable radially, but in the same relation circumferentially, and by placing former and cam blank on their appropriate face-plates with the pins entering the respective holes, the correct relation of the two is insured.

Figure 21 shows a face cam and its former in position in the milling machine, which is a Garvin profiling machine with a cam-cutting attachment. The slots in the face-plates which carry the pins entering the drilled holes in former and blank are plainly shown. The former and cam blank are geared together so as to rotate in unison, the former having a roller of the same size as the milling cutter and future cam roller pressed against it by a weight at the right, hung from the chain shown. Both roller and cutter are carried by the profiler head, and as the former and blank rotate it is obvious that the outline of the former will be reproduced on the cam. Several cuts are required to mill the groove, first to get the depth and then to complete the width — the finishing cutter of course taking a very light cut.

Retrospect

Looking back, it is clear that the cam was laid out in the position which it has to occupy when the crank begins its movement and the horizontal zero was marked upon it. The templet copied the drawing and transferred it and its zero to the former. The cam blank was keyed to the shaft when its zero was marked upon it and the marking was done with the crank located in its zero position, and these positions must be repeated when the parts are reassembled. Finally, the position of the zero of the former as determined from the drawing was made to agree with the zero of the cam blank as taken when in the machine. This being done with all the cams, it is plain that they must go to place and work as intended.

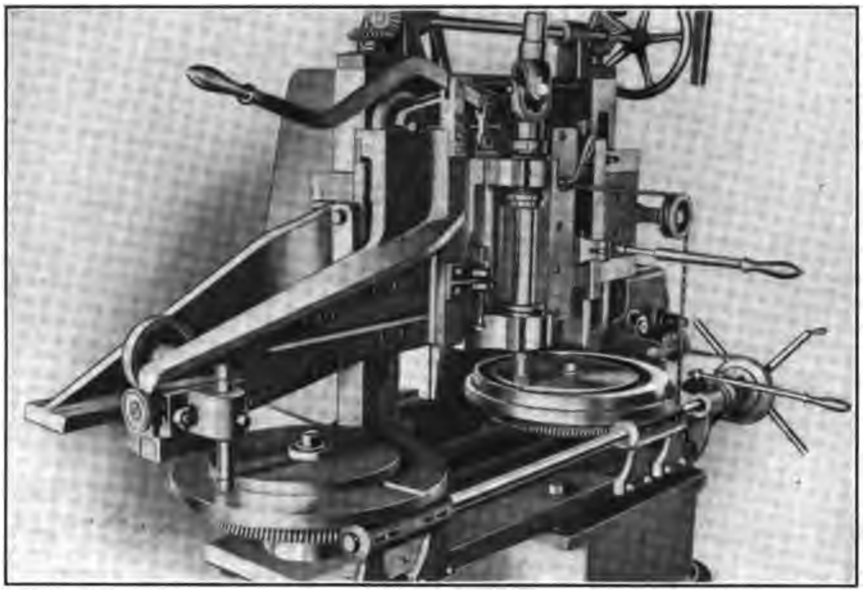

FIG. 21 Milling a Face Cam

CHAPTER VI

LAYING OUT AND MAKING PERIPHERY CAMS

THE laying out of a periphery cam cannot be done quite so directly as that of a face cam, as the layout must be a development and not a projection drawing. After laying out the profile it is transferred to a sheet zinc templet in the same manner as a face cam, which is then wrapped around the former on which the outline is scribed as in the case of a face cam. To provide for any minute discrepancy between the length of the templet and the circumference of the former, it is important that the place selected for the joint in the templet be within a straight part of the groove. Were it within an inclined part the result of such discrepancy would be a jog when the templet is wrapped around the former, but by selecting a straight portion this is avoided.

To illustrate the laying out of a periphery cam, cam 5 *winders wind link* of the chart has been selected and is shown in Fig. 22. To make the horizontal measurements on this drawing, it is necessary to translate the positions given in the chart by degrees into inches of circumference and, dividing the entire circumference of the chart by the number of five degree divisions, we find that for each one of them we must measure .785 inch when laying out the cams. The division point of the templet in this case is selected 4 inches to the left of the root of the curve of the chart, which nearly, though not exactly, coincides with the 180-degree line of the chart.

The chart informs us that the movement of the roller takes place during twelve 5-degree intervals, and 12 × .785 = 9.42 inches, or nearly enough for the present purpose, 9⅜ inches. It should be remarked that when evening up decimal into binary divisions the next smaller division should in general be selected, as the result is to slightly quicken the action which cannot in general lead to conflict. Laying down 9⅜ inches on the base line and measuring vertically the movement of the roller, 3½ inches, as shown on the chart, we have the rectangle within which the curve is to be laid down, and to find it we have only to follow the method given in Fig. 5. The throw circle is drawn and divided as before. Arcs *a b* and *c d* are drawn through the corners of the rectangle with the length of the radius arm as a center, giving the centers *e f*. The line *e f* is then divided and the intermediate arcs are drawn, the intersections of which with the parallels through the divisions of the throw circle give points on the pitch line of the groove from which circles with a radius equal to that of the roller define the groove. The dwell before the return movement takes place is then laid down and the return curve is drawn in the same way.

The displacement of the true zero from that assumed when laying out the chart is to be found, as was that of the face cam, by measuring on the general drawing the angular distance from the horizontal center line of the cam shaft to the position of the roller, and the position so found is then marked on the drawing and on the chart.

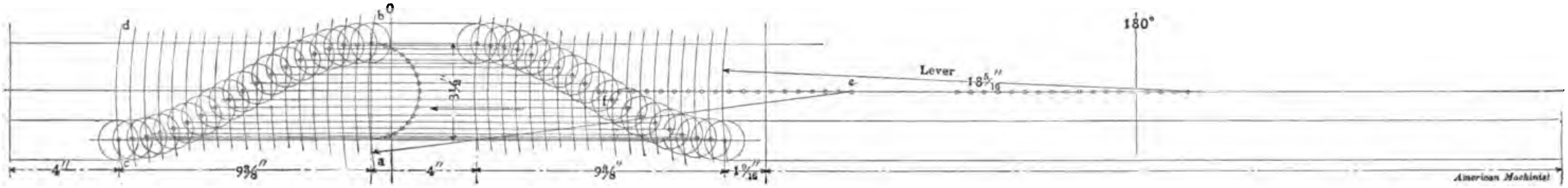

FIG. 22 Laying out a Periphery Cam

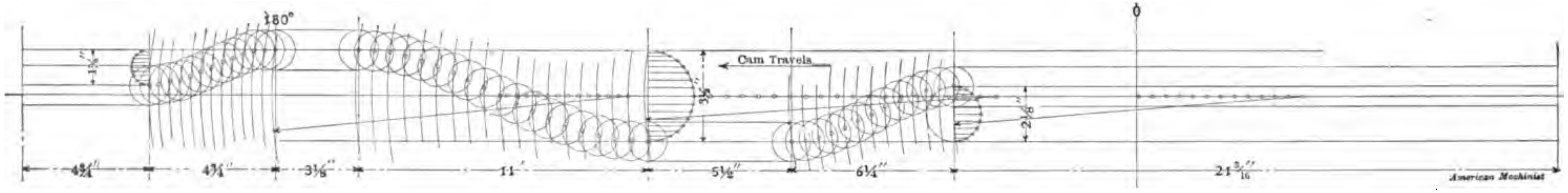

FIG. 23 Laying out a Two-step Periphery Cam

Figure 23 is given without detailed description to illustrate the laying out of a cam which makes the movement in one direction in two steps and in the return direction in one. It shows No. 14 *dial slide cam* of the chart. The diameter of the return throw circle will be seen to be equal to the sum of the diameters of the acting movement throw circles.

Locating the Zero of the Former

The zinc templet is laid out from the drawing in a manner not essentially different from that used with face cams, the zero being properly marked. Since, however, the former and cam of the periphery style are mounted on the same arbor when milling the groove, a different method is used to make their zeros agree. The pin-holes drilled through face cam and former serve to secure this agreement with face cams, but with the periphery style key-seats in former, blank and arbor serve the same purpose.

To secure the agreement of the key-seats and zeros, the formers are key-seated before they are scribed, and then mounted in the machine, with the keys in place, alongside their cam blanks, when the latter are rigged up for marking their zeros. The zeros on both blank and former are then marked at the same time. The parts being removed, the templet is then wrapped on the former blank, the zeros being made to coincide, and the former has the outline of the templet scribed upon it. After dressing the former to the scribed outline it is then placed on the key-seated milling machine arbor with the cam blank, and both are keyed thereto, the result being of necessity a perfect agreement between the zeros of former and blank.

The milling of the groove in the cam is shown in Fig. 24. It is done in the same machine as the milling of the face cam, but with a different fixture, and no detailed description of it is necessary. About the base of the machine are grouped the formers for all of the cams, both face and periphery, which belong to the chain machine.

Radius Milling Attachment

While the style of fixture shown in Fig. 24 is the one commonly used, and while it is sufficiently accurate for most purposes, it is not correct in principle, and in some cases, especially when the radius arm of the roller is short, another form of fixture becomes necessary. The error due to the fixture shown in Fig. 24 is shown in Fig. 25, from which it will be seen that at all points, except *a* and *b*, the roller is not directly above the center line of the shaft, and of course the groove should, by rights, be milled to correspond. The fixture shown gives a groove which looks directly toward the center line at all points and this leads to a tendency for the roller to cramp in the groove.

Figure 26 shows a radius milling fixture designed on correct principles, though not for show purposes, as it was gotten up hastily for a hurry job. The former is shown at the right and the cam being milled at the left. Roller and cutter, which in this case are conical, are mounted on radius arms of which the length is equal to the arm on which the roller is to be mounted in the complete machine, and the cutter is connected to the milling machine spindle by a universal jointed and telescopic shaft. The chain and weight for maintaining contact are not shown.

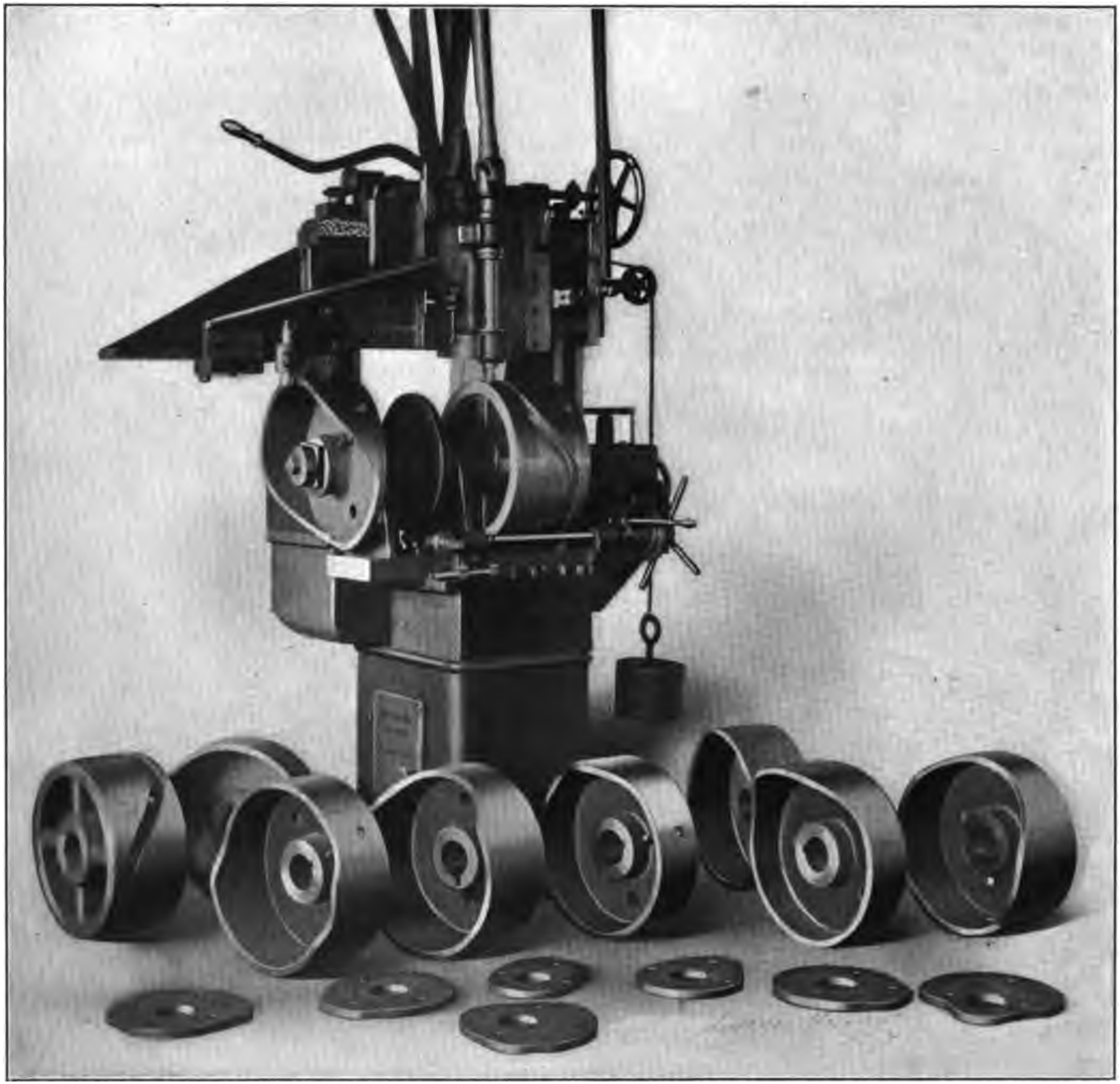

FIG. 24 Milling a Periphery Cam

Points to be Observed with Periphery Cams

Another feature of periphery cams which should not be overlooked is the dividing up of the arc of motion, which also is shown in Fig. 25. The lever is here laid out, as it should be, with the arc of motion divided by both horizontal and vertical center lines.

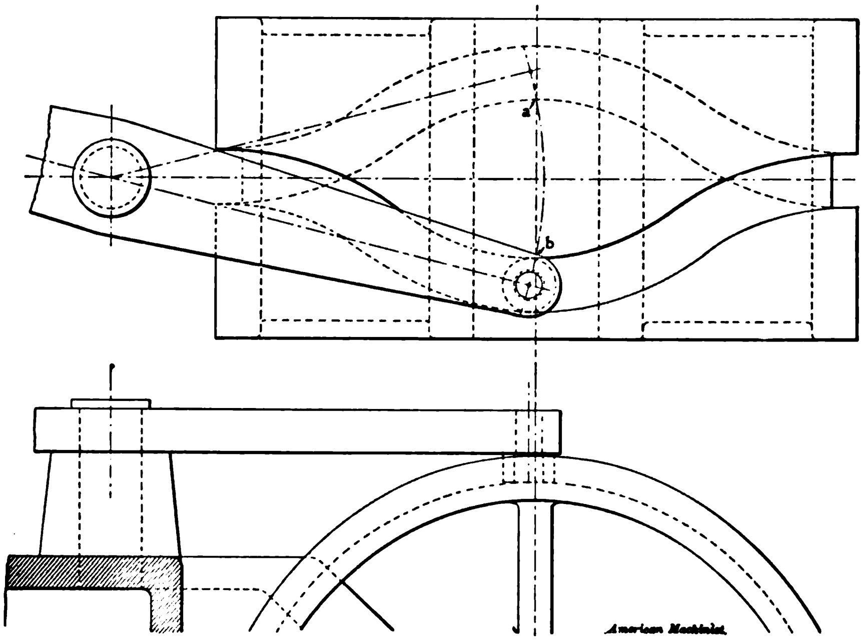

FIG. 25 Arc of Motion Properly Divided

Many designers make and recommend conical rollers for periphery cams, the rollers being laid out as are the pitch lines of bevel gears. This form of roller has, however, an end thrust which cramps the lever and leads to wear under heavy loads, and hence straight rolls are preferable. Of course their rolling action on the side of the groove is not theoretically perfect, and to reduce this the rollers should be made shorter than they often are. A length not much in excess of half the diameter is recommended.

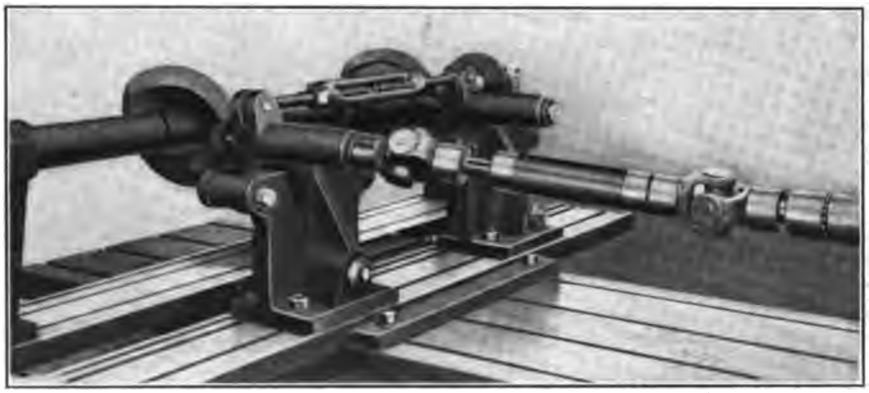

FIG. 26 Radius Milling Fixture

Another matter, which is of greater importance than would at first sight be thought, is to have the cam run away from the fulcrum of the lever instead of toward it, as the action is much the smoother with the former arrangement.

CHAPTER VII

RAISED PATHWAY, YOKE AND CONICAL CAMS

FIGURES 27, 28, 29, and 30 show raised pathway cams with two rollers which are used when it is desirable to be able to take up wear — the rollers being mounted on a block which permits their adjustment toward one another. Figs. 27 and 28 show respectively the profiles for periphery and face cams for operating a roller mounted on a slide, and Figs. 29 and 30 for operating a roller mounted on a radius arm, the four forms being shown in order to illustrate the effect of this construction in distorting the curves, whereby, unlike the two edges of the grooves heretofore shown, the two sides of the ridges are no longer parallel and in some of the forms they are very far from parallel.

The pitch lines of cams of this style are laid out as in the previous cases, the centers of the rolls being found by striking arcs *a a a*, *b b b*, which intersect the lines of movement of the slide or the arcs drawn from the division points of the radius arm center line.

Successful cutting of this style of cams for use with rollers mounted on a radius arm can only be done on a radius arm attachment similar in principle to that shown in Fig. 26, the length of the radius arm of the attachment being equal to that which is to carry the roller. Moreover, this attachment should have two cutter spindles so that both sides of the ridge may be cut at the same time. A single-sided former with one roller bearing against it is all that is necessary in that respect, but the side chosen should be adhered to. For periphery cams it is immaterial which side is chosen, but for face cams the outer curve is preferable, as it is of larger diameter and has easier curves and thus produces more satisfactory work.

YOKE CAMS

Figure 31 shows an excellent form of cam which is sometimes available. As in all other cases, the first considerations are the throw and the angle of the cam's movement during which the movement of the rollers takes place. The movement may be either continuous or intermittent, the latter case being shown. Radial lines *A B*, *A B* are drawn to include the angle of movement, which is then divided as in former cases to agree with the divisions of the throw circle, and the curve of the opposite side is then so laid out that the distance between the diagonally opposite rollers is, in all cases, the same.

The yoke may obviously be placed on a radius arm, in which case the radius lines become arcs of circles struck from the circle through the center of the fulcrum pin.

CONICAL CAMS

There is a great variety of special forms, of cams which are occasionally required, and of these Figs. 32 and 33 show a conical form used recently in connection with a paper-

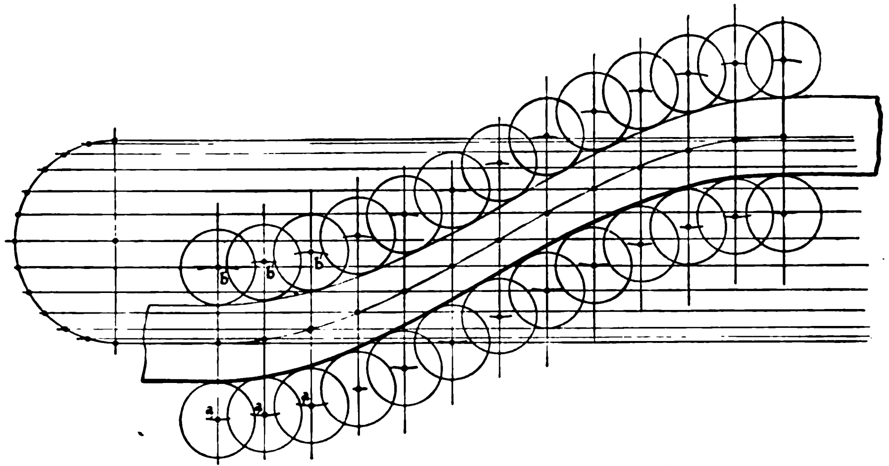

FIG. 27 Periphery Cam for Operating a Slide

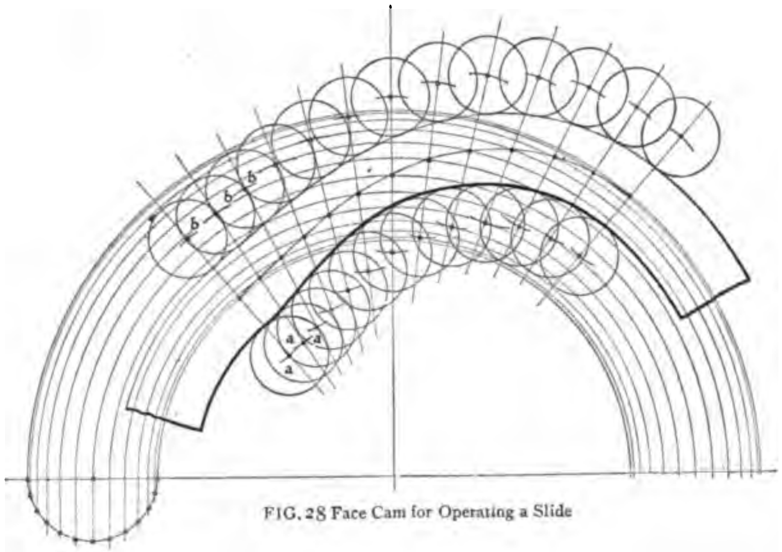

FIG. 28 Face Cam for Operating a Slide

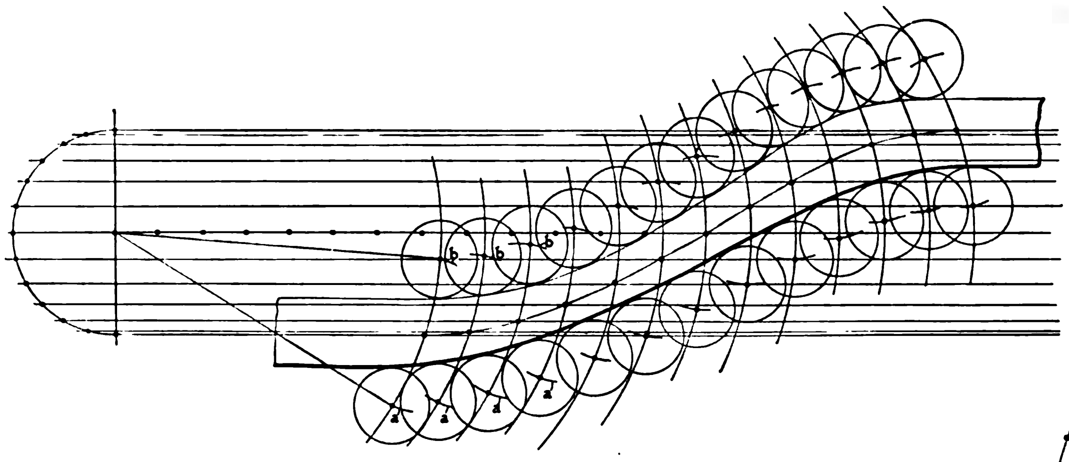

FIG. 29 Periphery Cam for Operating a Pivoted Lever

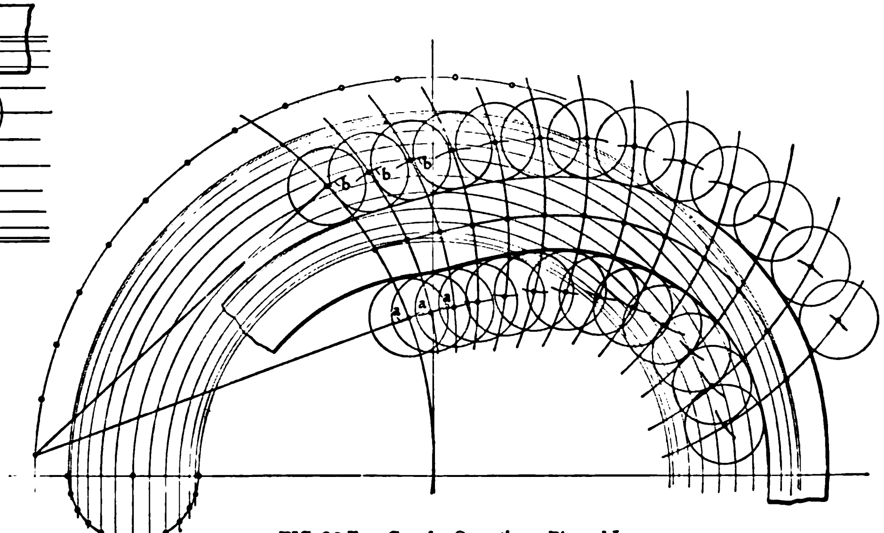

FIG. 30 Face Cam for Operating a Pivoted Lever

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RAISED PATHWAY CAMS WITH TWO ROLLERS

folding machine. In Fig. 32 the groove is external to the cam and the roller is mounted on a slide, while in Fig. 33 the groove is internal and the roller is carried by a radius arm. The angle of the cone is of course determined by the connections to the operating parts. while the diameter, as with other forms, is made such as to keep down the angle of the groove.

In laying out this form, the length *a b*, Fig. 32, of the side of the cone is taken in the dividers and the arc *c d e* is drawn with it as a radius. The length of this arc is then made equal to the entire circumference of the base circle of the cone *b f* and the arc *c d e* is then to be treated as though it were a complete circumference of 360 degrees and the angles *g h i* and *j h k* laid down to cover such parts of the arc *c d e* as the actual arcs of movement cover parts of the circle of the cam. The throw circle and the profile of the groove are then laid down as in previous cases and the templet is cut, the ends being defined by the radii *c h* and *e h*. The templet will then wrap around the former, the end lines meeting as they should.

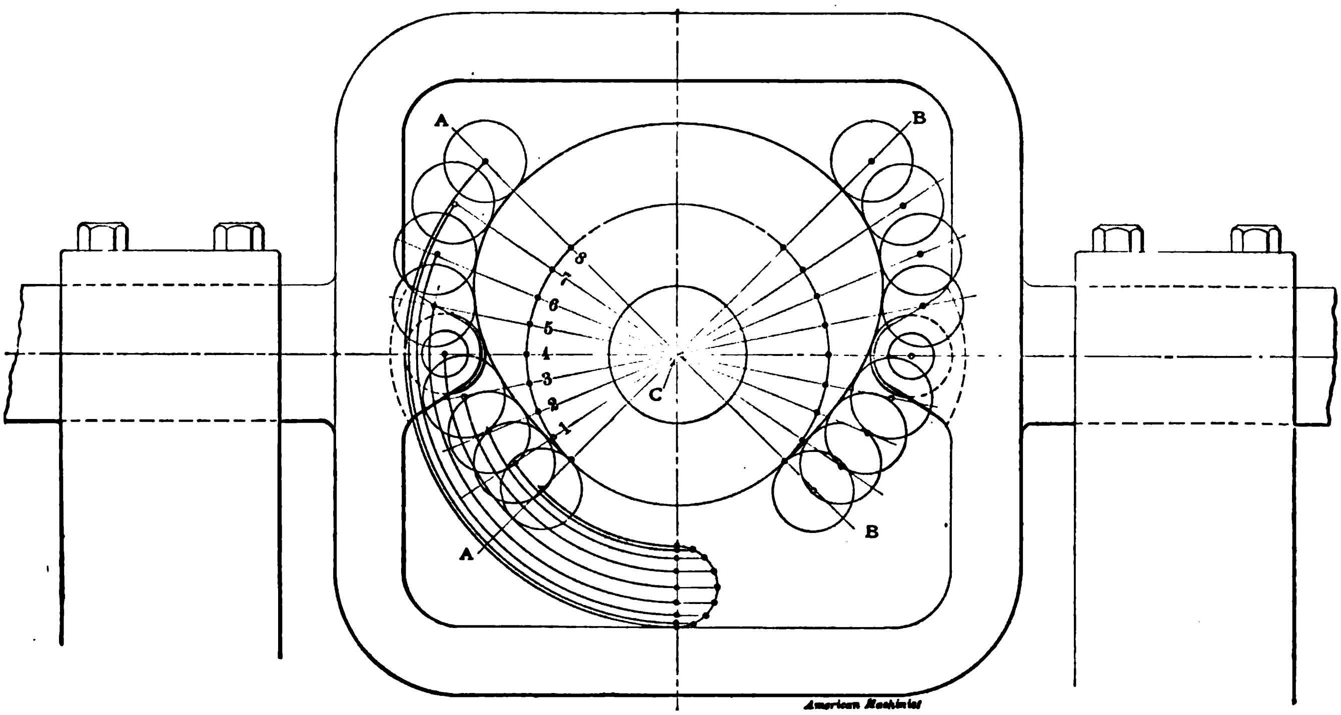

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FIG. 31 A Yoke Cam

The circle *l m* is equal in diameter to the pitch circle of the cam groove at its outer radius, and points *n, o, p, q* of the templet fall on *r, s, t, m*, respectively, of the former, the arcs *n o*, *o p*, and *p q* being equal respectively in length to *r s*, *s t*, and *t m* of the former.

The procedure of Fig. 33 will be understood without lengthy explanation after what has gone before.

The length of the radius arm *a b* is taken in the dividers and the circle *c d e* is found such that the arc *f g* struck from it with the length of the arm as a radius shall pass through the center of the cam shaft, and the cam curve is laid down precisely as with face cams.

Figure 34 shows a fixture for milling conical cams. The former is shown at the left,

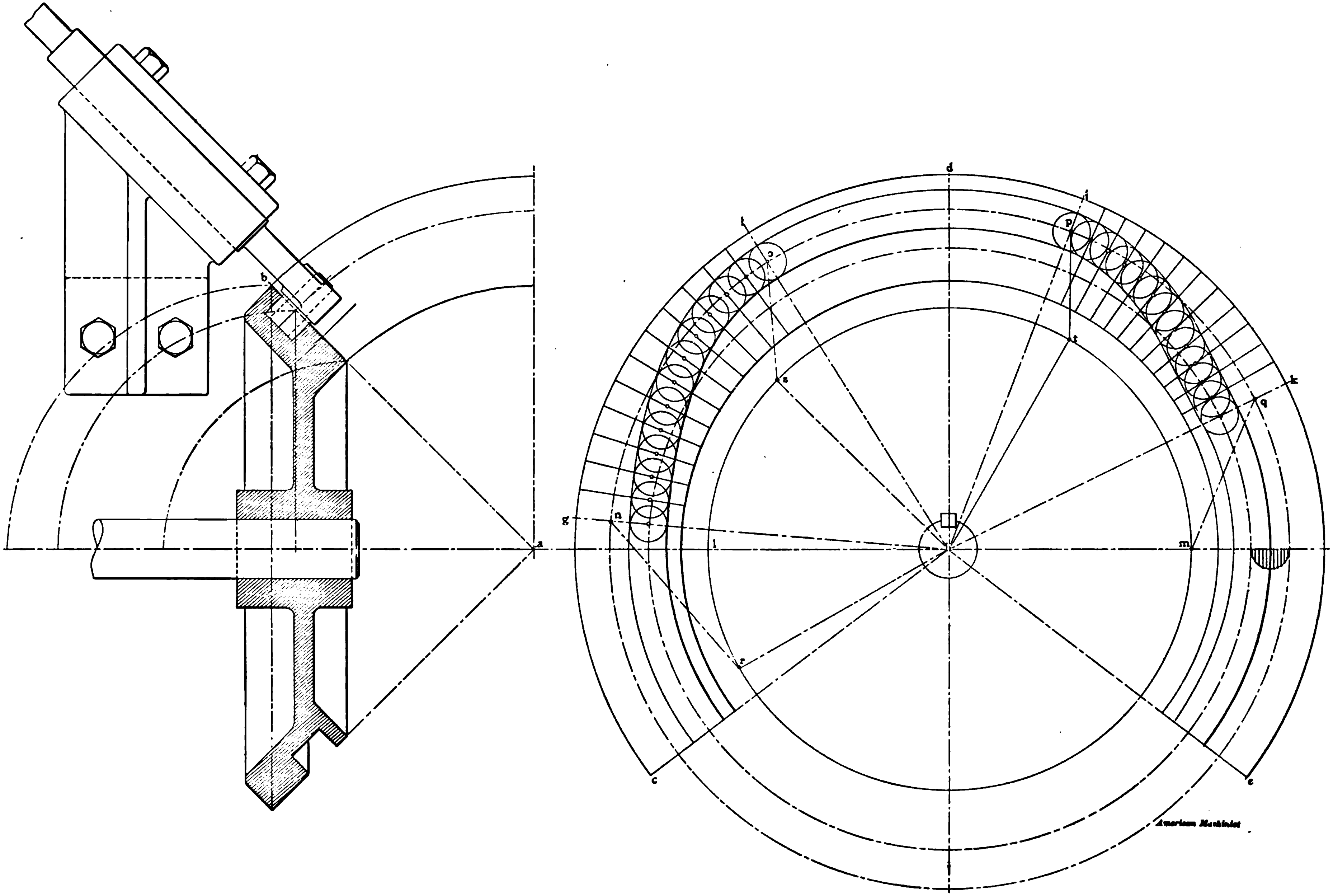

FIG. 32 A Conical Cam Operating a Slide

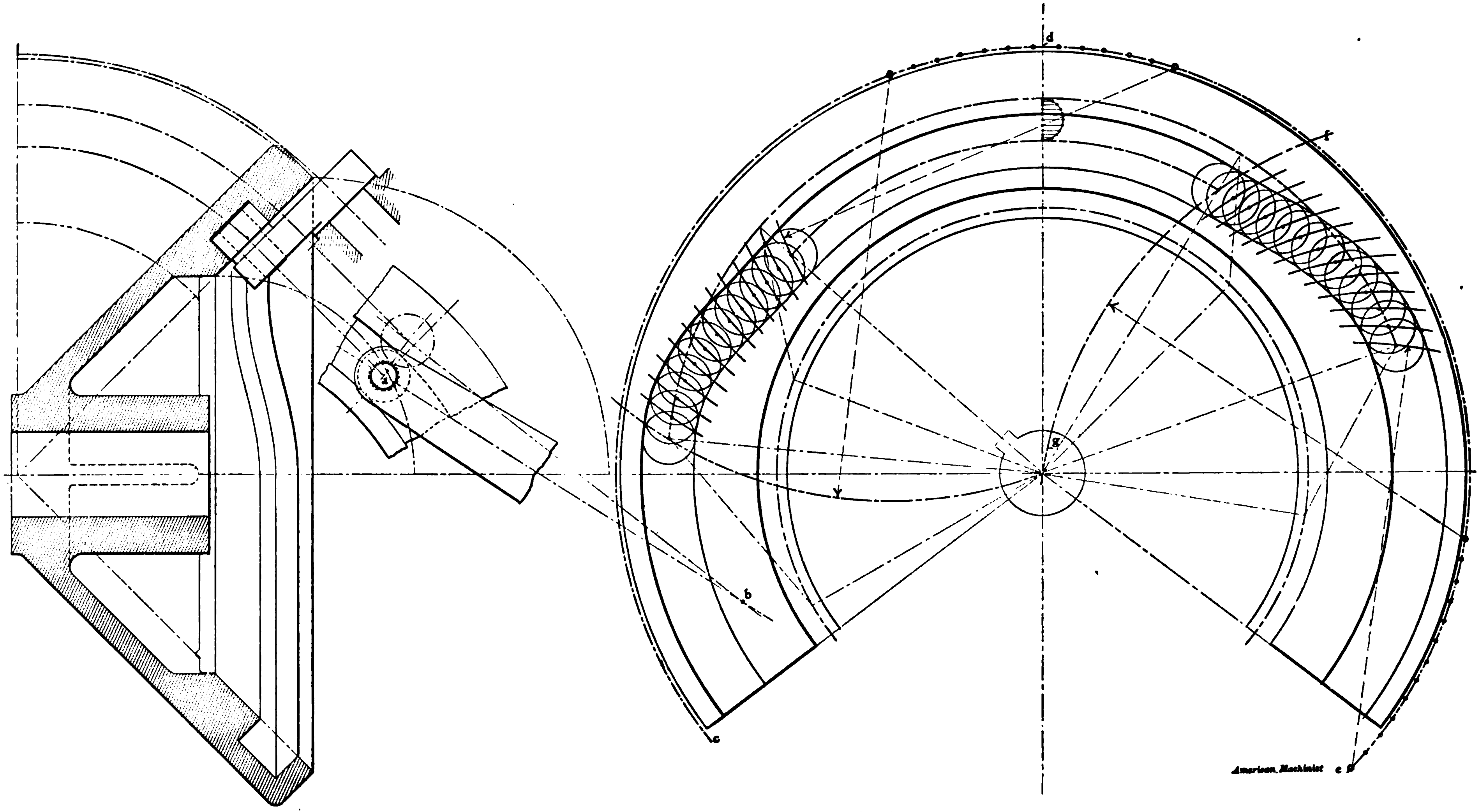

FIG. 33 A Conical Cam Operating a Radius Arm

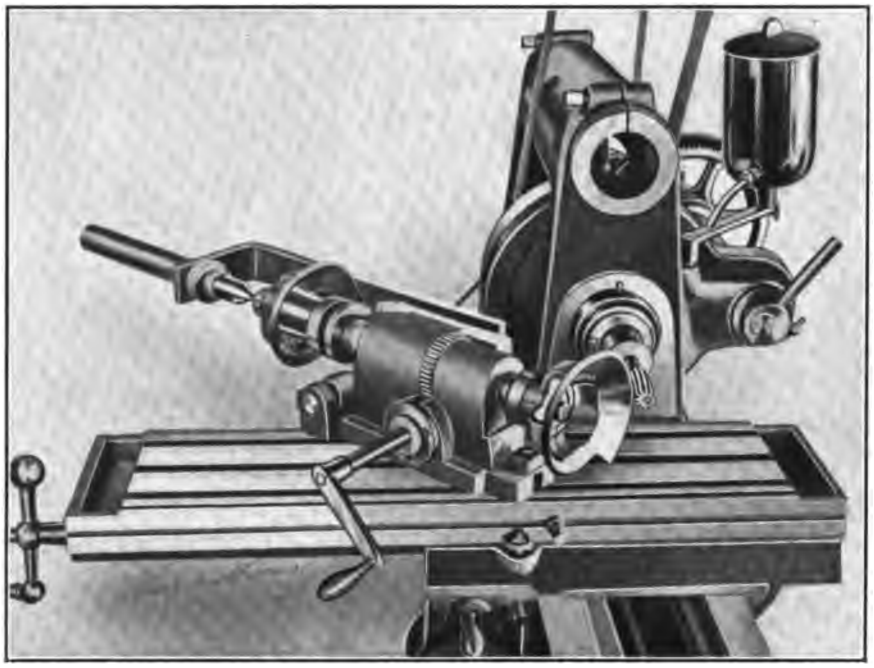

FIG. 34 Fixture for Milling Conical Cams

its stop, which represents the roller, being behind the former and having its edge parallel with the milling cutter and rounded to the same radius. Its length, together with that of the cutter, must be such as to reach the inner portions of the cam. Of course a roller might be used here as in former cases, but with this cam it was not done, as the cam had a spring return and hence but one acting edge. Contact of the former with the stop was maintained by a spring within the tube at the left, this spring acting upon a long center and causing the former to follow up the stop.

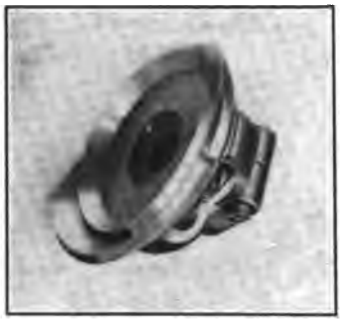

FIG. 35 Adjustable Conical Cam

This particular cam is of additional interest, as the beginning and ending of its movements are adjustable, a feature which may be used with spring return but not with cams which give a positive movement in both directions. Fig. 35 shows it to be in two parts, of which the circular portions or dwells overlap, one part carrying the beginning and the other the ending of the movement. The two parts are secured to the shaft and to one another by split grips and bolts, as shown.

CHAPTER VIII

CAMS FOR PRESCRIBED MOVEMENTS

THE general principles from which cams for prescribed movements are laid out may be gathered from the methods given in Figs. 6, 7, 10, and 11, for cams giving a uniform instead of a crank motion, for the uniform motion is often prescribed and hence those examples fall under this class. The general method is to work backward from the movement required and thus determine the form of the cam groove which will give the movement instead of laying out the groove to give an easy start and stop.

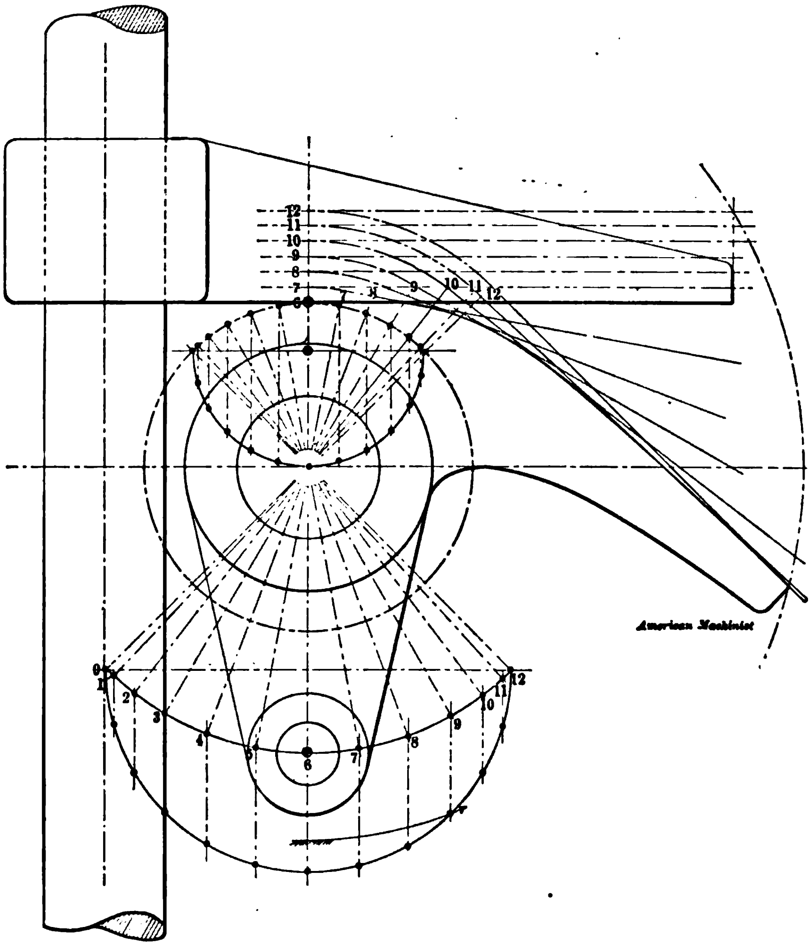

FIG. 36 A Prescribed Movement Cam

Figure 36 is another illustration of this method, which is given purely as such, and not as an example of the methods followed in steamboat engine work, in which this general form of cam is chiefly used.

The vertical arm is driven by an eccentric which of course revolves uniformly, the angle of swing of the arm being 90 degrees, as shown by the arc of movement. The chord of this arc is made the diameter of the throw circle, which also represents the eccentric movement and which is divided in the manner already repeatedly shown, and the division points are projected to the arc of movement and numbered. Radius lines through these points give the corresponding positions of the vertical arm and these radius lines are carried past the center, as shown.

The toe is supposed to be raised from its seat during half a circle of revolution of the eccentric, and to be at rest on its seat during the other half. If the movement of the toe is to be uniform, its throw is divided into equal parts to the same number as the divisions of the throw circle, and arcs are struck from the center of the wiper shaft and through these division points. The radius lines extended to cut these arcs give points from which tangents are to be drawn, and a smooth curve tangent to these tangents is the required curve of the wiper. It is obvious that any required movement could be given to the wiper by properly spacing the division points of its lift.

A Combination Movement

Of all prescribed movements the uniform motion is the most common, and it is frequently used in combination with the crank motion, the latter being used to give an easy start and bring the parts up to the speed of the uniform movement and again to gradually slow down the movement at the end. An example of this is given in Fig. 37, which is from the paper-folding machine already referred to.

A is a roll over which the paper is brought to the working parts of the machine. A shield *C* starting from rest at position *D* must at *E*, where it picks up a flap in the end of the paper, reach the speed of the roll and thereafter travel at the same speed as the roll and deliver the end of the paper between the feed rolls *a b*, the end of the movement being on line *F*, when the shield must return to its initial position. The size of drum *A* is such that the shield must make two complete movements for each revolution of the drum.

The angle of the drum movement during which the initial movement of the shield is to be made is 24½ degrees, and the angle during which the uniform movement must be made is 38 degrees, both as shown. The drum and cam shafts are connected by gears *A*, *B*, *G* to give two revolutions of the latter to one of the former, and angles of 49 and 76 degrees — twice those of the drum shaft — are laid down as shown. Radial lines *c d*, *c e*, *c f* at these angles determine the position of the beginning and ending of the two cam curves. The movements of the sector pinion *H* for the two movements of the shield are determined and laid down on the pitch line of the sector *I*, each side of the center line of the sector's movements, and radius lines to the center of the sector arm determine the two throws of the roller on arc *J*. Dividing the first angle of movement into equal parts, we draw the arcs 1 *a*, 2 *a* , 3 *a*, etc., through them and then lay down the throw circle *K*, which is here a quadrant only, and from divisions obtained as before draw arcs, of which 0 *a* and 6 *a* only are shown, and thus determine the first portion of the cam curve. The second uniform portion of the movement is laid down in equal divisions at the left of the

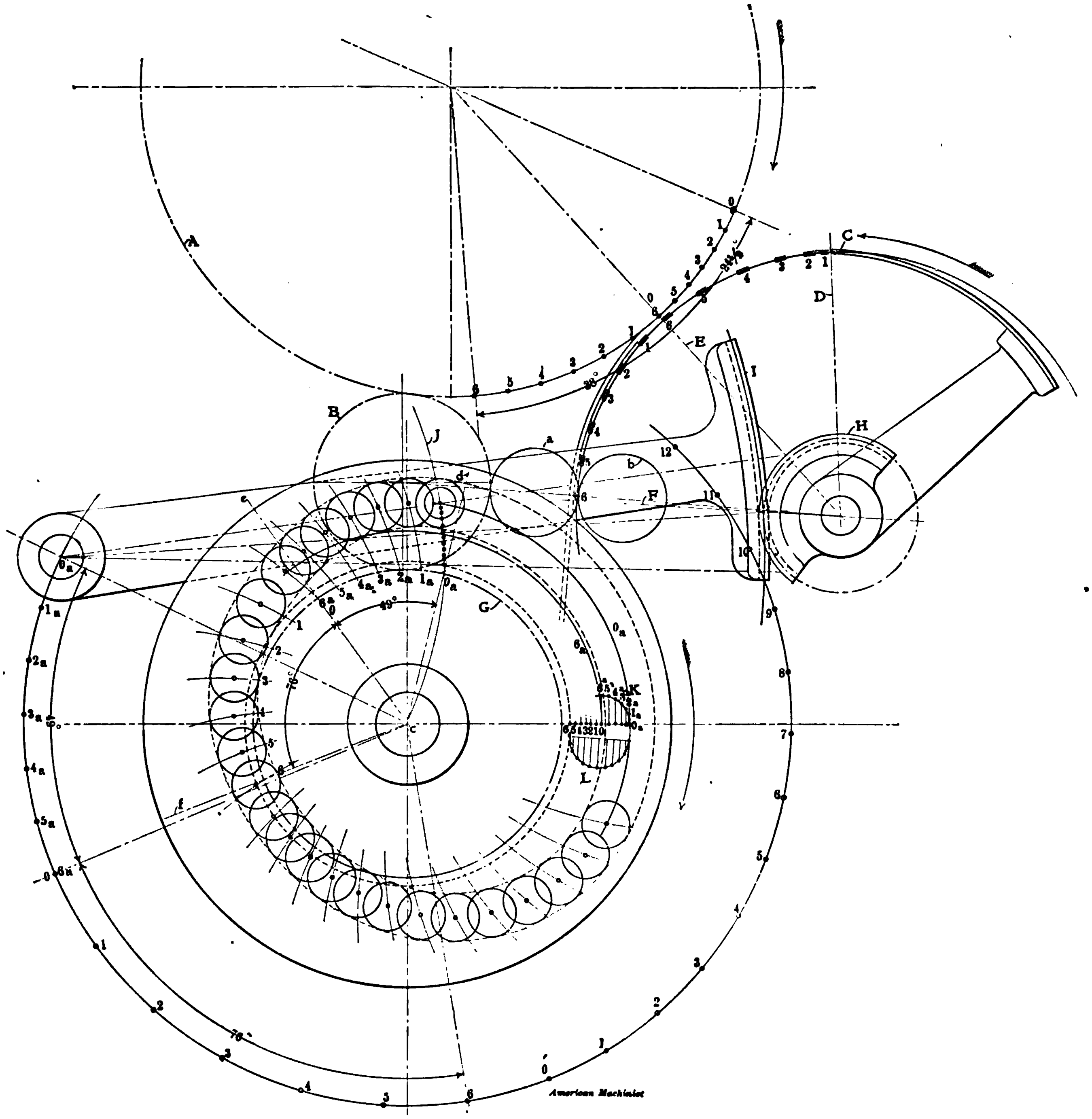

FIG. 37 A Combination Cam

throw circle, and arcs through these divisions determine the second portion of the movement. The complete return is laid out from another throw circle *L* and the beginning of the first curve and ending of the second are connected by arcs of circles from the center of the cam shaft and represent a dwell in the movement of the shield. The curve as thus determined is, however, slightly modified by easing off the junctions of the uniform and crank motion curves in order to avoid abrupt corners.

The above example will serve also to show the method of dealing with cases in which cams are mounted on shafts that run at different speeds, the angles of the main driver being multiplied or divided, as the case may be, when laying out the various cams. It is often an advantage to speed up certain cam shafts, as thereby the number of degrees during which movements take place being increased the necessary size of the cams is correspondingly reduced. In this way cams which might otherwise be of unwieldy size may often be reduced to acceptable dimensions.

DETERMINATION OF THE PATH OF MOVEMENT BY CAMS

In the above cases of prescribed movement the operating points were guided in circles or straight lines by radius arms and slides, the requirement as to the cam relating to the movement only. It frequently happens that not only must the cam determine the speed of the operating point, but it must also determine the path of the movement, and such cases often develop the fine art of cam design.

Such a case is shown in Figs. 38 and 39, which show a cam movement from actual practise, and it will serve, without other examples, to show the method of dealing with cases of this kind.

A lever *A* is supported and driven by a radius arm *B* and a slide *C*, which are actuated by cams *D* and *E*, the requirement being that the end *F* of lever *A* shall move in a circle and at uniform speed.

The method of laying out the cams is shown in Fig. 39. The required circle *P* is drawn and divided into equal parts, and trial lengths of the levers and positions of the fulcrums are assumed and laid down. From fulcrum *G* draw the arcs *H* and *I*, the latter being divided equally by the horizontal and vertical center lines. Find the extreme positions of lever *A*, one of which it will be observed is not on the horizontal center line of the circle of movement of *F*. These positions determine the throw of the slide *C* which carries the pivot *J* through which lever *A* passes. The slide is driven by the lever *K* on which the roller is mounted. From the fulcrum of this lever draw arcs *O L* and *M*, and cut the former from the centers of the extreme throw of slide *C*, and thus determine the extreme positions of the roller. Now take the distance *F N* in the tram, and, starting at point *F*, cut the arc *I* for each division point of circle *P*, marking corresponding points with numbers and letters to avoid future confusion. When all the points have been found on arc *I* draw a line from each division point on circle *P* through the correspondingly numbered points on arc *I*, cutting the center line of the slide *C*, and number the points to agree with the previous numbers. Set the tram to the distance of *J O*, cut the arc *O L* and draw radii to the roller lever fulcrum. The intersections of these radius lines with the arc on which the roller moves with the arcs struck from the circle *P* through the fulcrum center and equally spaced around it, in the same number as the divisions of circle *P*, give the profile of the center line of the disk cam.

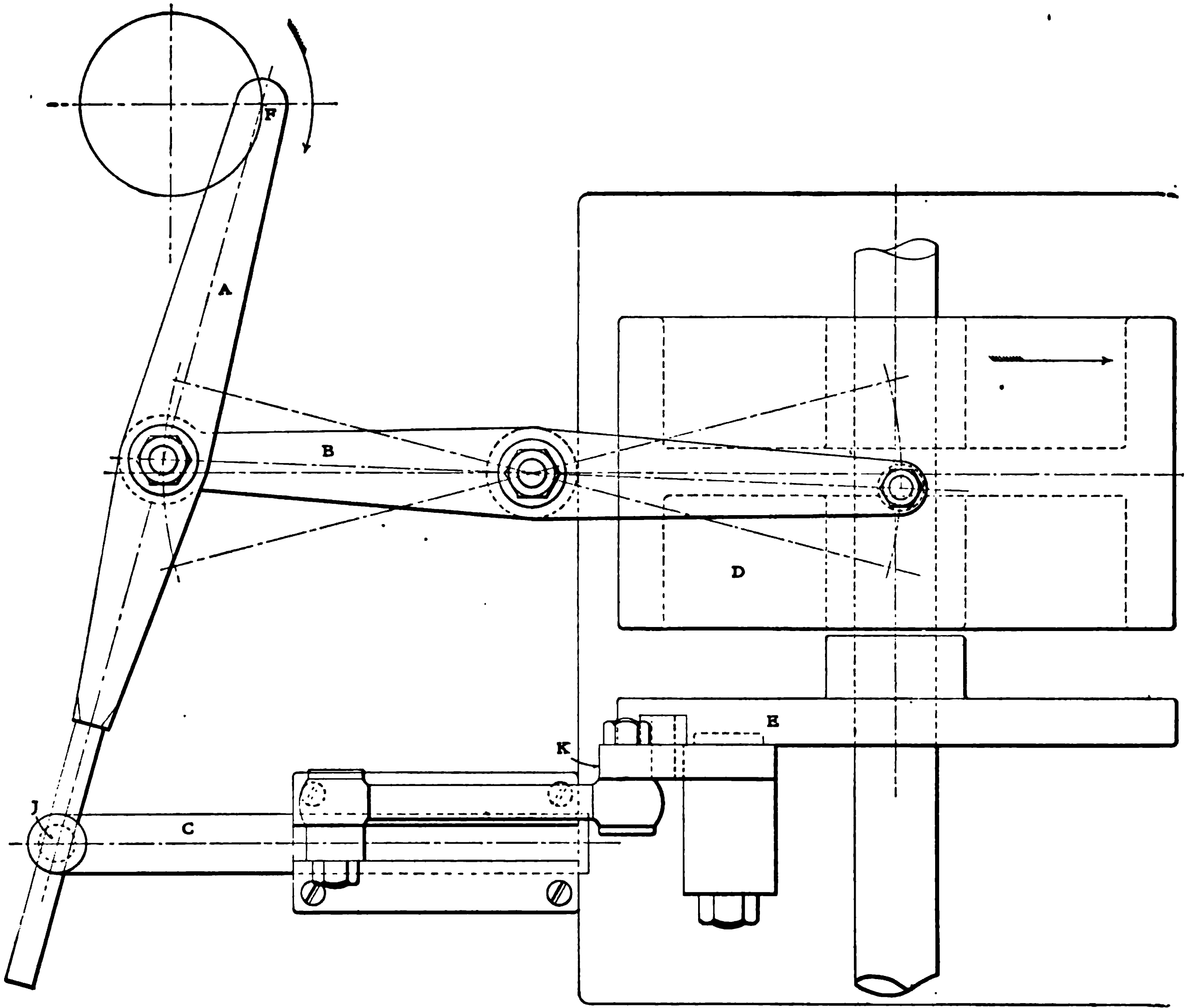

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FIG. 38 Path of Movement Determined by Cams

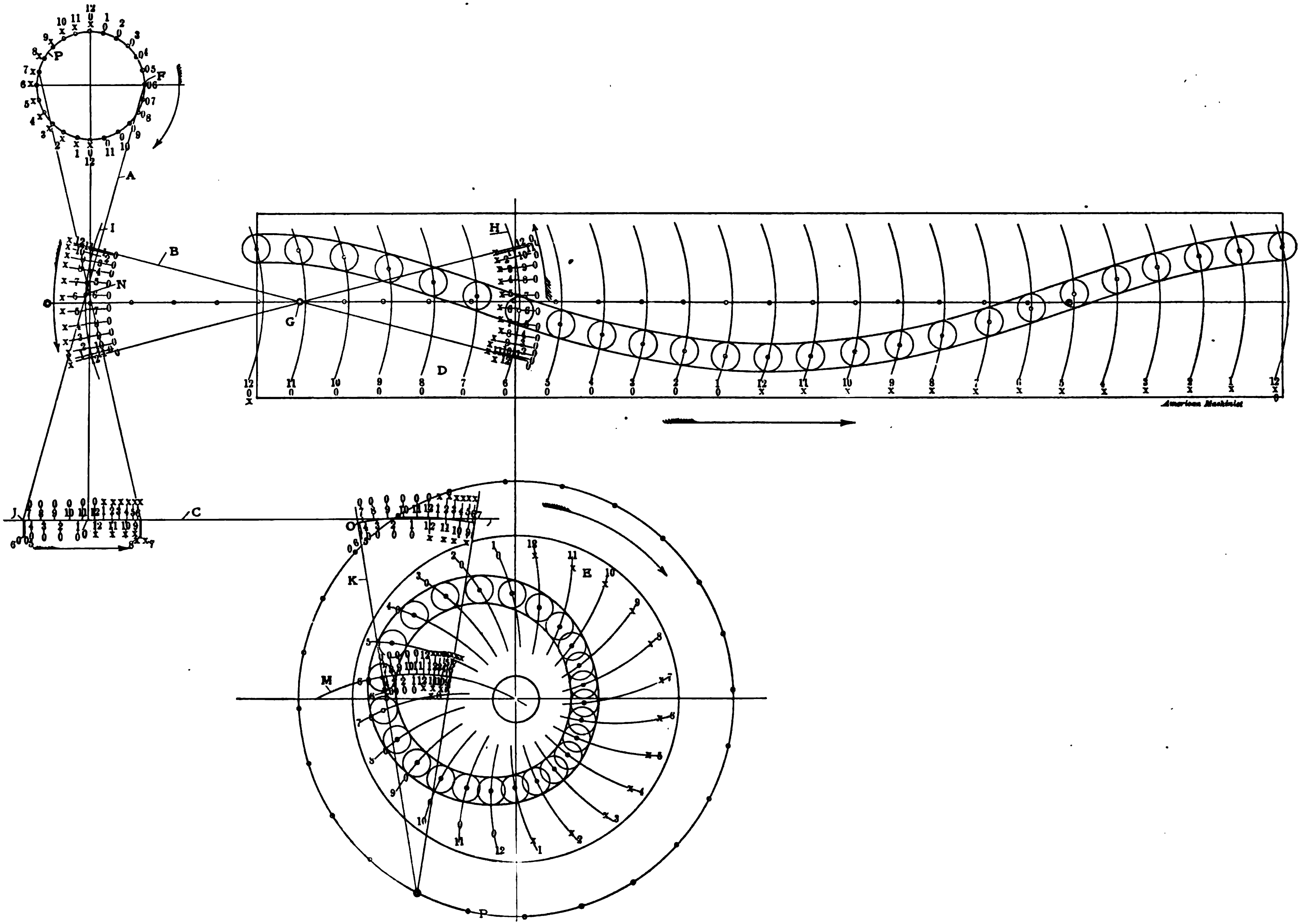

FIG. 39 Laying out the Case of Fig. 38

To find the roll centers of the periphery cam which drives lever *B*, draw lines from the division points on arc *I* through the fulcrum *G* cutting arc *H*. When these points have been established and numbered assume the circumference of the cam, divide it into the same number of parts as those of circle *P*. Draw arcs from fulcrum center *G* with a radius equal to that of the roller arm, and intersections as shown define the cam groove.

The movement shown is continuous, but it might be made intermittent if desired, though the cams would have to be larger to maintain equally favorable angles, and, obviously, the prescribed movement might be a square or any other figure within reason.

Figure 40 illustrates a special cam which gives a square movement in a very simple manner, the movement obtained being shown by the square at the left.

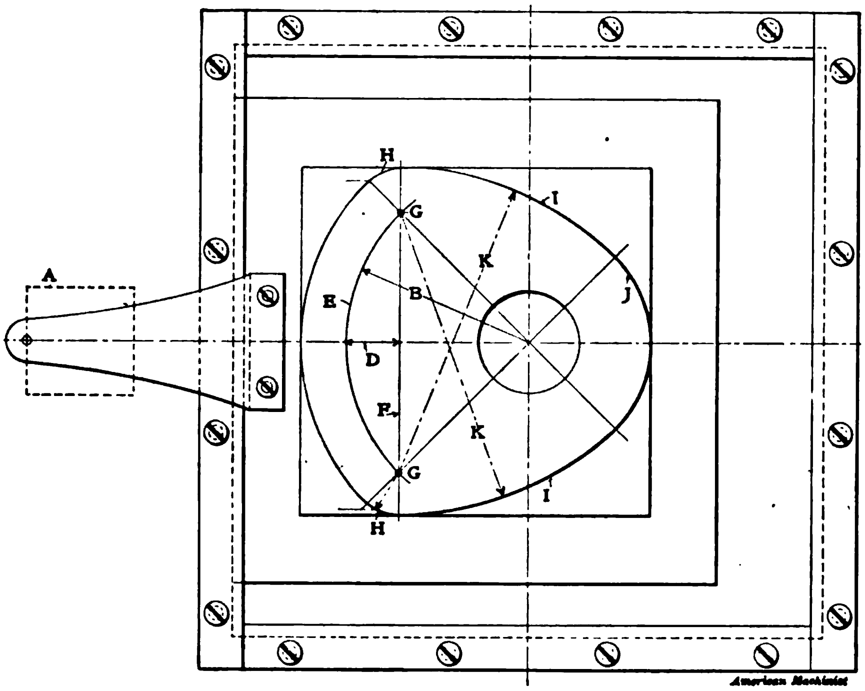

FIG. 40 A Square Motion Cam

To lay out this cam the radius *B* is assumed sufficient for the movement required, and the arc *E* is drawn. From the intersection of this arc with the horizontal center line the length *D* equal to half the required throw is laid down, and the perpendicular *F* is drawn. From the intersections *G G* the arcs *H H* are drawn, and from the same centers, with radii *K K*, the arcs *I I* are drawn. Finally from the center of the shaft the arc *J* is drawn, completing the outline.

This cam may be made to operate two surrounding frames placed one above the other, the cam being as thick as both frames combined. If the opening of one of these frames is increased horizontally and the other vertically, and the frames are properly guided, a horizontal movement will be given to one and a vertical movement to the other, the movements being alternate. By connecting one of the frames to the operating point by a bell crank two parallel slides may be given alternate movements, and by varying the length of the bell-crank arms the movements may be made to differ in length.

CHAPTER IX

REPEATED AND RETURN MOVEMENT CAMS

FIGURE 41 shows what may be described as an intermittently operating worm gear. The cam is of the raised pathway type, and has two complete turns of the pathway on its periphery. The driven member may be a disk having a series of rollers projecting radially from its circumference, or, better, from its face. The illustration will make it plain that these rollers will engage the cam groove in succession, the disk being turned by a series of steps with long dwells between.

Figure 42 shows a cam in which two revolutions are required to effect the complete going and return movement, and in which these movements take place at the same point in the revolution of the cam shaft, although this latter feature may be easily changed. In this construction, the usual roller is replaced by a boat-shaped piece *a*, which is swiveled on a supporting pin and is of sufficient length to insure its following the cam groove in the desired manner. It is obvious that at each revolution of the cam the boat will reverse its position and take the opposite groove to the one followed during the previous revolution.

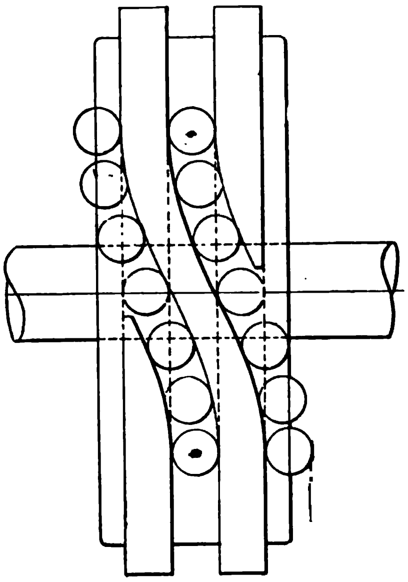

FIG. 41 Intermittently Operating Worm

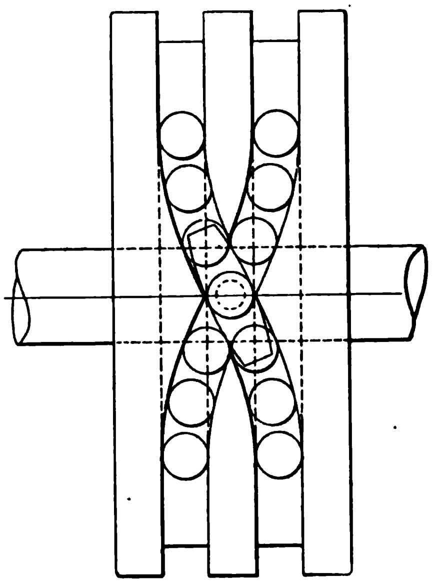

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FIG. 42 Two-Revolution Cam

Figure 43 shows a cam for which a patent was at first refused, the examiner claiming that it would not work; but in face of the fact that it had been made and did work, the patent was subsequently allowed. Its object was to stop the action of a feed plunger at predetermined intervals, the movement of the plunger being effected by another cam not shown. The cam is, in a sense, a combination and modification of those shown in Figs. 41 and 42. It gives a step-by-step motion, as in the case of Fig. 41, and reverses this motion as in the case of Fig. 42; but the reversal, instead of being on the succeeding revolution, as in the case of Fig. 42, may be made to take place after any desired number of revolutions.

The cam *a* contains a switch *b*, which is let into a recess, as shown. It is held in place by a ring *c*, and has at its inner end a series of teeth which are engaged by the rack *d*. At each revolution of the cam the die *e* is turned the width of a tooth, and at the completion of the movement one or the other of dogs *f g* engages the rack, pushes it endwise and reverses the switch; thus, on the succeeding revolution, reversing the motion of *e*. A spring-latch *h*, engaging notches in the switch, holds it in place. The dogs *f g* were mounted in a T-slot, and the reversal could thus be effected after any desired number of revolutions of the cam.

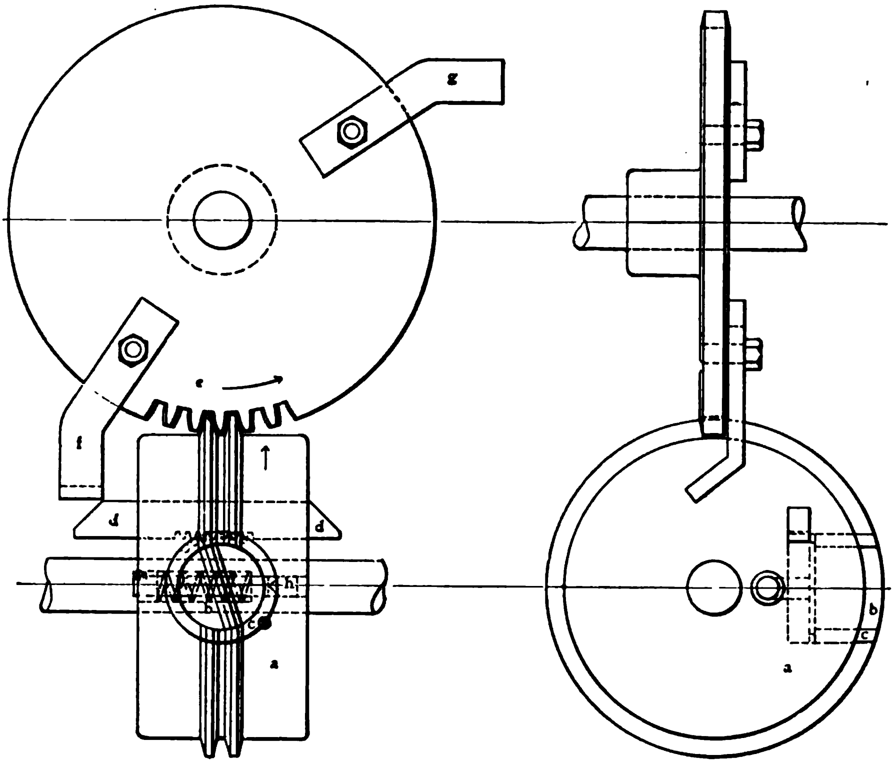

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FIG. 43 Reversing Cam

This is one of the rare reversing mechanisms that do not involve the use of springs, weights, or momentum to complete the throw-over, and this is due to the fact that the throw-over is accomplished by the movement of the driving instead of the driven piece. The movement of the driven piece merely locates the position of the step *f* or *g*, the actual reversal of the switch being by the movement of *a* while *e* is at rest.

As shown, the device is not mechanically perfect, as regards the action of the teeth on the cam surfaces, and the teeth are limited in thickness and must be rounded to enable them to engage properly. Nevertheless, the arrangement worked satisfactorily. With a sufficiently coarse pitch, the gear could be made in the form of a disk, with properly shaped rollers for teeth, and this construction would, no doubt, be better than the one shown.

CHAPTER X

CHARTS WITH A SEPARATE BASE LINE FOR EACH CAM, ABBREVIATED CHARTS, EXTREME ANGLES, LOCATING KEYWAYS.[1]

I DO not approve of all of Mr. Smith's methods and will state my objections categorically.

LAYING OUT THE CHART WITH CAMS OVERLAPPING

I very much prefer to give each cam a separate base line and space all its own, as in Fig. 44. In this way the whole path of motion of any cam is seen at a glance, while to pick out a path from the chart shown in Fig. 18 is a job for a novice, and any one is apt to lose it while hunting for another path to compare with it. In this way the numbering scheme, which is new to me, could be much more clearly carried through. Further, I like to see the motion of the real working-point and throw of the cam both dimensioned on the chart, as shown in Figs. 44, 45, and 46. This practise will save a good deal of trouble and turning over of tracings. It will be seen that curve marked *Feed* in Fig. 44 has a separate base line, but cuts through and encroaches on the space reserved for *Stroke of Gate*. This is a perfectly legitimate and sometimes very useful exception to the rule. I may as well add at the start that I constantly take liberties with all these, my self-appointed guides, as every one should who is master of his own.

STICKING TOO CLOSE TO THE ONE MACHINE FOR A TYPE OF CAM CHART, AND THUS GIVING THE IMPRESSION THAT IT MUST ALWAYS BE ELABORATE AND ACCORDING TO FIXED RULES.

For example, three weeks were spent on the chart mentioned. It is frequently necessary to have the complete set of drawings in the shop in that time. In designs where cam charts are necessary it rarely happens that more than two or three movements are so related that every point in their course must be known. Especially is this true in designs that are modifications or enlargements of existing types of machines. For these few the precise accurate path may be plotted on the chart and the remainder and larger number simply represented by straight lines. All that is necessary to know, *i.e.*, where they start and stop, is thus shown. When there are several motions on a machine from 8 to 10 inches in length, it is desirable to scale them down to 2 or 3 inches for sake of clearer representation; in all cases, of course, putting the proper dimensions in place. Figs. 45 and 46 illustrate my meaning in these particulars. In curve, *Nut Holder Out and In*, Fig. 45, the only portion plotted accurately is at *A*, where it is necessary to know just when

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[1] By Roy H. Smith.

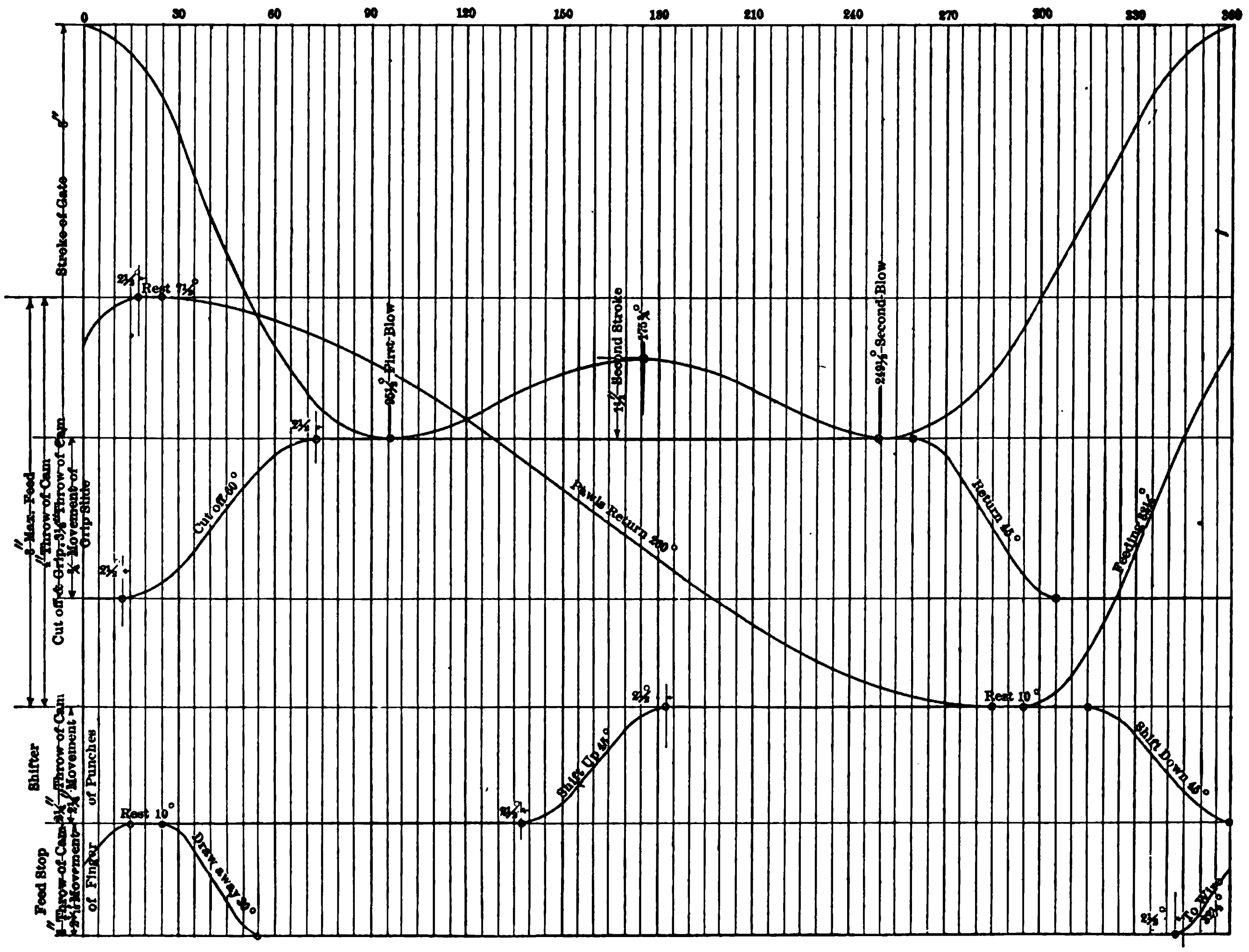


FIG. 44 Cam Chart with Separate Base Lines

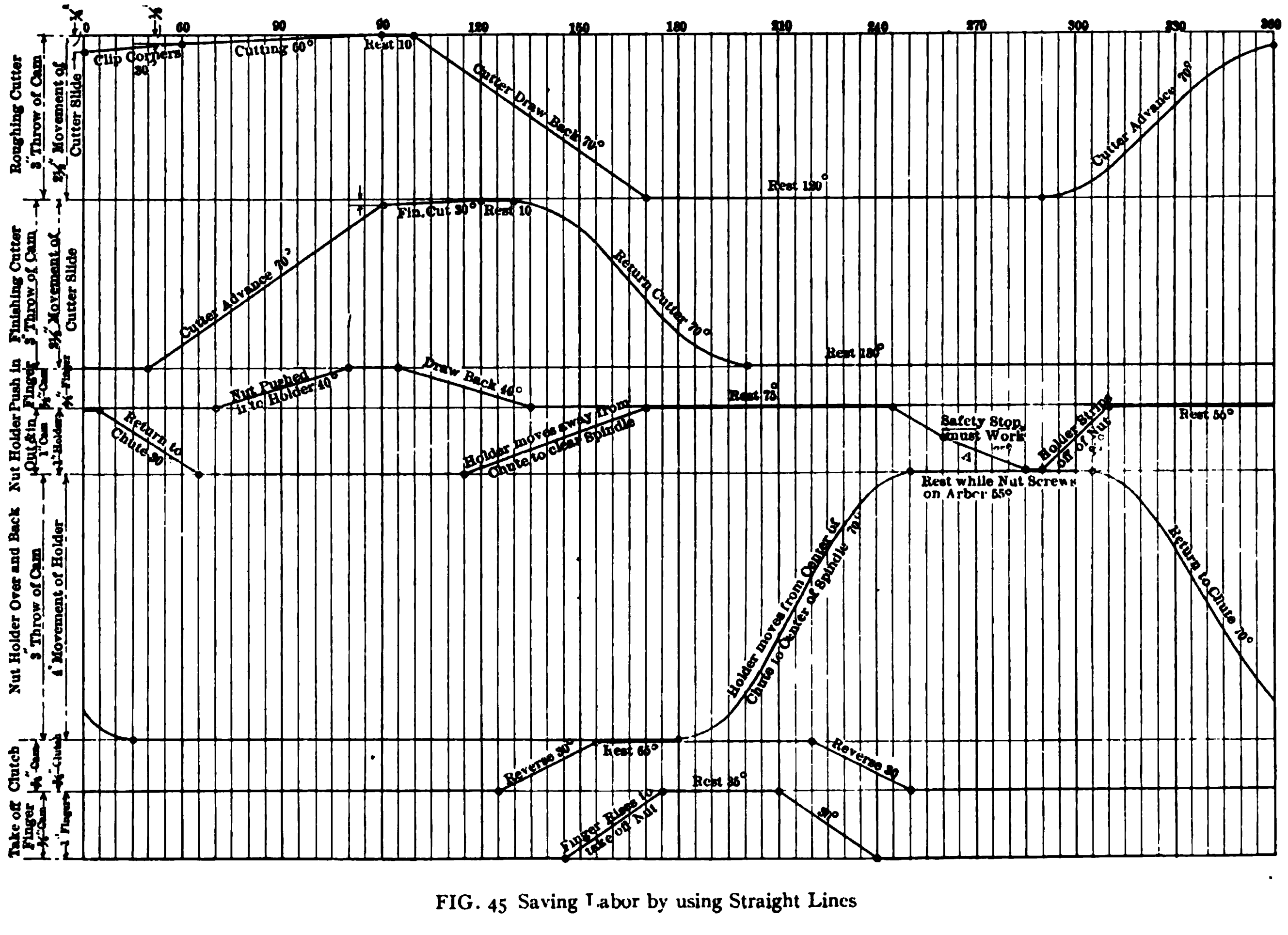


FIG. 45 Saving Labor by using Straight Lines

the safety must act. The only other curves whose paths it was desirable to know on this machine are shown as plotted curves; other times are indicated perfectly as well, and with less effort and time, by straight lines.

CAM ANGLES

While it is very desirable to keep the cam path down to an angle of 30 degrees or less, it frequently cannot be done, and sometimes, on rapid-running machinery, to increase the diameter of the cam to make the path gentler would lead to more serious troubles. I have seen in use for a good many years reciprocating slide cams with an angle of 60 degrees. The rollers on these start gently, get up to speed and run up a straight 60-degree incline to the top, *i.e.*, about ½ inch of the straight face. About 130 reciprocations are made per minute. They work hard, of course, but they work commercially, and that is enough.

THE METHOD OF LOCATING KEYWAYS IS SLOW, EXPENSIVE, AND UNNECESSARY

A great many cams simply have the path cast on and are smoothed up with a file. With these the keyway must be located by means of the chart. Large heavy duty cams, with rolls, say, from 3 inches diameter by 1½ inches face to 6 inches diameter by 3 inches face, are usually made of cast-iron with inserted tool-steel faces where the hard work comes. It is necessary to core out for all these, and again the location of the keyway is important. The same is true where the complete cam is made of tool-steel, and which must be forged to shape, for it is too expensive to cut it away. In my own work I frequently meet with cams weighing from 200 to 1,000 pounds, most of them with inserted steel faces and many with gear teeth on the periphery. Manifestly no system of assembling or trying out could be tolerated with these, for either keyway or gear-teeth location. I follow with uniform success Mr. Smith's method for locating the zeros of individual cams. Of course no system of checking this part of the work can be too tedious or laborious if it is accurate. Sometimes, for greater safety, I make paper models of the cam outlines and actually try them in place on the drawing.

To go back to Figs. 45 and 46 for a moment, we have here an excellent example of the value of careful study of the cam chart. Fig. 45 is the chart for a nut-finishing machine which had been running for years, finishing the nut, Fig. 47, where marked *Top*. This operation had some time ago been discontinued, and only the *bottom* was finished in this machine. After careful study it was found possible to do away with the finishing-cutter, trim a little here and there and push all the cams back closer together until the movements were all completed in 180 degrees of movement. From this it was only a step, the form of cams happily permitting, to lay out all the motions on the same cams again and get just double as many finished nuts as formerly — as shown in Fig. 46. It will be seen that neither the cutting-speed nor the feed has been altered. Of course an extra safety stop must be put in to stop the machine when a bad nut comes down.

OTHER FORMS OF CHART

Figure 48 shows a form of chart sometimes met with. It represents exactly what Fig. 44 represents, except that the degree lines are arranged in a circle, but it is not to be

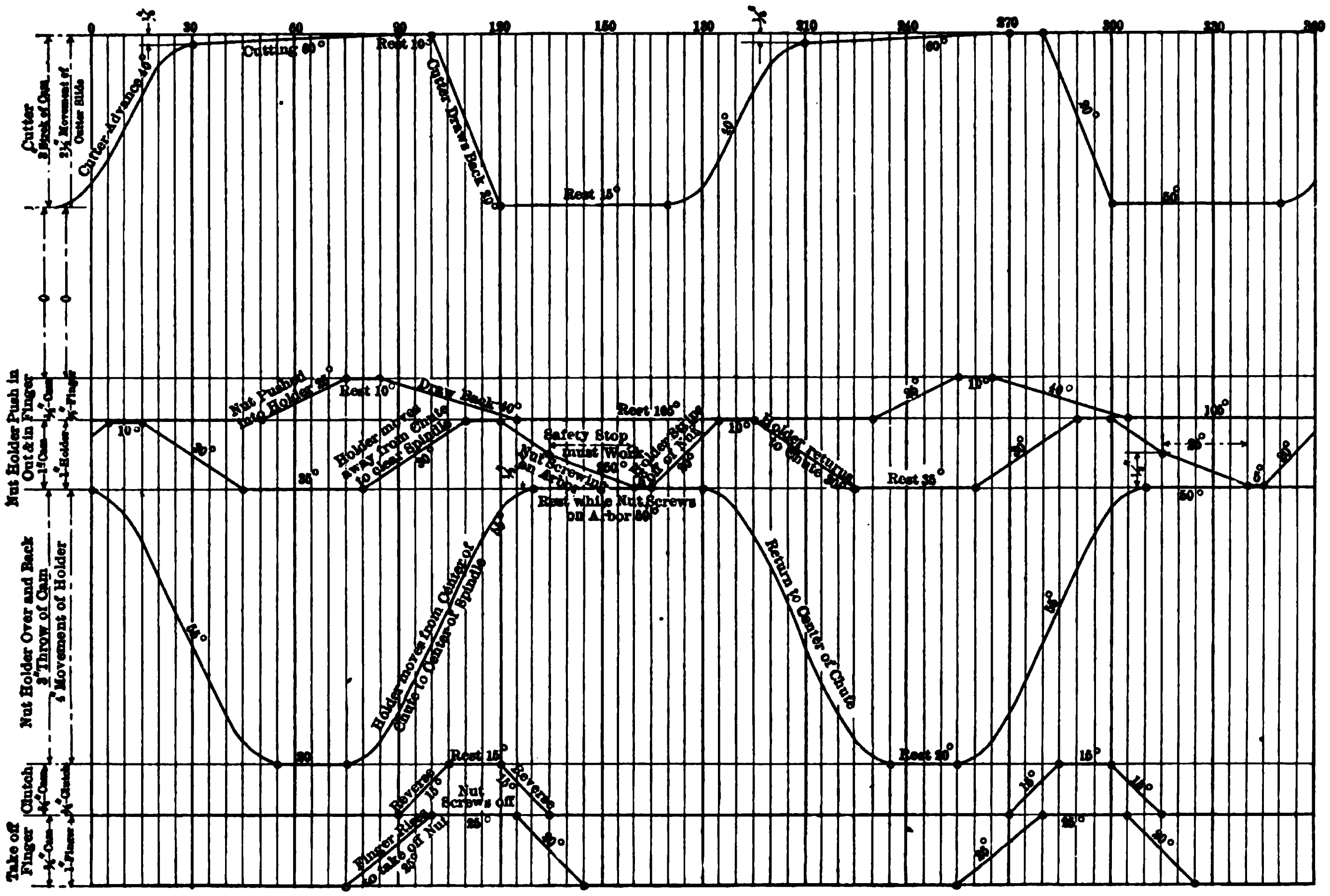


FIG. 46 The Chart of Fig. 45 Rearranged to Double the Output

compared with the other style of chart for simplicity. There is still another style of chart I have met with a good deal. Fig. 48 would illustrate this method if each curve, instead of starting from a common zero, were to start from its own individual actual zero, *i.e.*, line of motion. This is a laborious and unwieldy chart, but helpful in locating keyways.

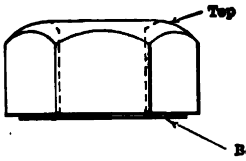


FIG. 47 Work done on Machine Chartered in Figs. 45 and 46

At page 31 Mr. Smith makes a statement that is either misleading or not "far-leading" enough. He is speaking of a yoke working in ways (Fig. 31), and adds, "The yoke may obviously be placed on a radius arm." Of course it is clear enough to him, but I think

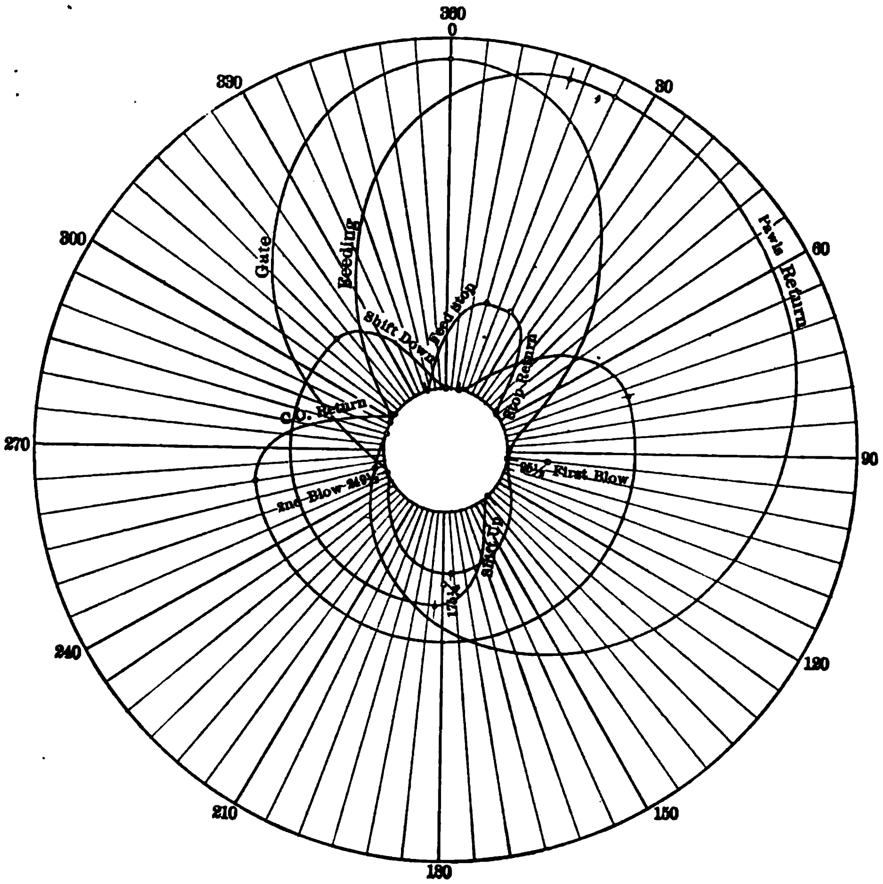


FIG. 48 An Undesirable Form of Chart

it important enough to put in black and white, that a yoke on a swinging arm is quite a different thing from a yoke sliding in ways. A yoke like Fig. 31, sliding in ways, can only go back in the same time that it goes forward — a yoke on a swinging arm can never do this.

CHAPTER XI

THE DOUBLE-CAM SYSTEM OF THE MONOTYPE

As originally made, the cams of the Monotype were of the face variety, with a raceway for the roller; that is, both acting and return movements were positive. In use these cams proved to be short lived, both roller and raceway wearing badly, so badly, in fact, that with some of the cams the usual life did not exceed six months, and this with hardened rollers and pins and with steel pieces inserted in the cams at the places where the greatest wear developed. On investigation the conclusion was reached that this was due to the resistance offered by the inertia of the roller to reversal on its pin, for, with this style of cam, such reversal accompanies each change of the roller from the inner to the outer surface of the groove and *vice versa.* While this action is always present with this style of cam, it was accentuated into a serious matter in this case by the unusual speed of the cam shaft — 140 revolutions per minute — and the resulting inertia developed in the rollers.

In order to overcome the difficulty the cam system shown in Fig. 49 was devised, in which a pair of conjugate cams, *a* and *b*, are keyed to a pair of shafts which are so geared as to run at the same speed and in the same direction as indicated by the arrows. The roller *c* lies between them and is driven in one direction by one cam and in the opposite by the other — the two cams being formed by methods to be described, which insure the absence of slack between them and the roller in all positions. It will be observed that with this arrangement the direction of rotation of the roller on its pin is never changed. Its speed, of course, varies with the diameter of the cam surface acting at the moment, and to this extent its inertia comes into play to induce sliding, but such changes in speed are small in comparison with reversal and, moreover, they are always gradual, whereas the reversal with the usual style of cam is abrupt. So successful has this construction proven that the life of these cams is unknown. They have now been in use for six years and in no case have they shown serious wear except when subject to obvious neglect in oiling. In other words, they are at least twelve times as durable, under the conditions of the Monotype, as the old style, and this with the cams made of gray iron.

MODIFICATIONS OF CAM CURVES FOR HIGH SPEED

The inertia developed at the speed at which these cams run has given rise to other difficulties. At the start they were laid out from the crank-motion curve used by Mr. Smith, but modifications of this have been found necessary. At slow speed the acting cam will do as it is intended — push the roller the required distance — but at high speed the inertia of the driven parts comes into play in precisely the same manner as with the reciprocating parts of a high-speed steam engine. The parts are started from rest and their motion is accelerated to a maximum when they are slowed down to a stop. If the speed is high the inertia of the parts resists this stopping, precisely as the inertia of the reciprocating parts of an engine resists stopping. If this inertia exceeds the resistance which is offered to the movement of the parts, the roller leaves the acting cam and contacts with the return cam, which then stops it, precisely as the crank-pin of a high-speed engine stops the piston. The effect of this was to add the looseness of the joints to the intended movement of the parts, or, with resistance before the parts, to make the movement uncertain in extent at the very point where certainty was essential.

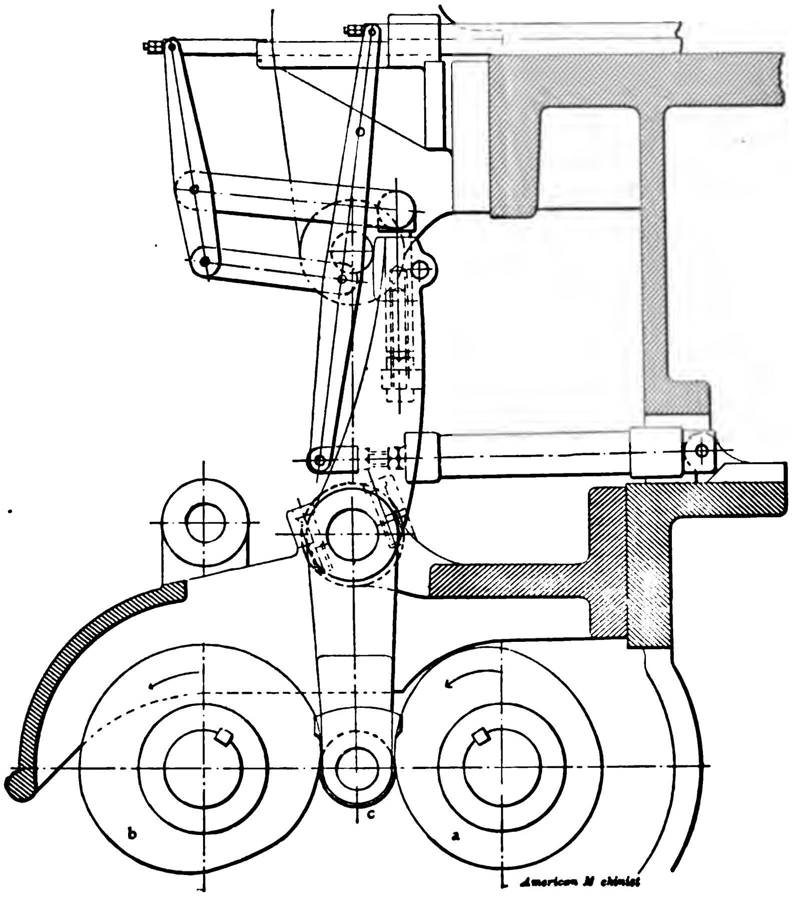


FIG. 49 The Monotype Cam System

The first remedy tried for this was to place springs before the parts to resist their movement, the idea being that the springs should be strong enough to more than overcome the inertia of the parts and thus cause the roller to keep contact with the acting cam throughout the acting movement, but it was found to be impracticable to use springs strong enough to accomplish this with the crank-motion curve, and the final resort was to a modification of that curve.

The difficulty arises solely in the stopping of the parts and, the time during which the cam may effect the movement being fixed, the remedy is primarily to reduce the time during which the parts are accelerated and to increase the time during which they are retarded. This is shown in Fig. 50 in which the chart curve for one of the cams is given. The parts reach their maximum speed while the cam turns from *a* to *b*, which points include but 25 degrees of motion. From *b* to *c* the chart curve is straight, while the stopping occupies the angle from *c* to *d*, or 50 degrees. The curves as laid out are not based upon any formal system, but are simple arcs of circles. The starting curve is struck from the center *e*, while the retarding curve is composed of two arcs struck from *f* and *g*, which latter, by accident, falls upon the circumference of the extreme cam pitch circle. By the use of two arcs in the retarding portion of the curve, the negative acceleration of the driven parts is reduced at their point of union *h*, its value at the end of the movement, which is the chief point in consideration, being thus materially less than if the entire arc *c d* were struck from a single center.

The location of point *h* is calculated with reference to the weight and velocity of the driven parts, and is so located that the energy of these parts, due to their velocity at point *h*, is small enough to be within the capacity of a practicable spring to absorb. By these means the difficulty has been entirely overcome. The return curve *i j* is formed in the same manner, the starting portion being struck from one center and the stopping portion from two, as shown. There is, however, no calculation of energy made for this curve, as it is unnecessary.

It will be observed that the steepest portions of the curve materially exceed the limiting angle of 30 degrees recommended in previous pages. The angle of *b c* with the base line measures with the protractor about 37 degrees.

It should not be understood that all the cams are laid out in this way. In some the weight of the parts or the required exactness of the movement is not such as to require it.

Laying Out the Cam Curve from the Chart

In the Monotype, levers having equal arms are seldom used, and Fig. 50 shows the method of making the reduction. The chart curve is laid out to give the movement of the driven piece. If the cam is to have half the throw of this piece, for example, the height *k l* is divided in half and the lines *l m* and *n m* are drawn. Points on the chart curve being then projected to the line *l m* and from the intersection down to *n m*, the heights of the last intersections above the base line give the distances to be used in laying out the pitch curve of the cam. Circles representing the extreme diameters of the acting cam are then calculated and drawn, the calculation being based on the distance between the cam shafts, the diameter of the roller, and the required movement. The various angular lines are then drawn and the height of the points on line *n m* are stepped off inside the outer circle. These points determined, their positions are approximated as closely as possible by a series of circular arcs. Thus the extreme diameter ends on the 95-degree line, as shown by the chart. The radius *o p*, struck from a point on the radius line for this angle, gives the next portion of the cam pitch curve which extends to the line *q r*. From a point *q* on this line, the next portion of the pitch curve is drawn extending to the line *q s*, and so on. The actual cam curve is, of course, drawn from the same centers, the radii being reduced by the radius of the roller.

The mating cam is laid out in the same manner, although this is not shown in the drawing. The curve for the mating cam is, however, used only for the purpose of getting out the pattern and to insure that the curve does not contain structurally impossible forms, such as concave arcs of less radius than the cam roller. Should such curves develop, the chart must be modified to remove them. The actual transfer of the curve to the iron former is made for the acting cam of each pair only, the curve of the return cam being developed from it, as will be described.

Outlining the Master Former

The transfer of the curve to the master former is made by the use of the iron drawing-board shown in Fig. 51. This board contains a pocket in which the master former blank is dropped. A stump, fitted with a key on which the blank is a close fit, locates it positively in position, and a protractor ruled on the surface of the board, but only imperfectly shown in the half-tone, enables the angular lines to be ruled on the surface of the blank, which is coppered for purposes of laying out. The profile of the drawing is then repeated with all possible care on the blank, the coppered surface permitting a good degree of accuracy, although in its main dimensions — the least and greatest radii which determine the movement — the accuracy is not dependent on this layout. The profile

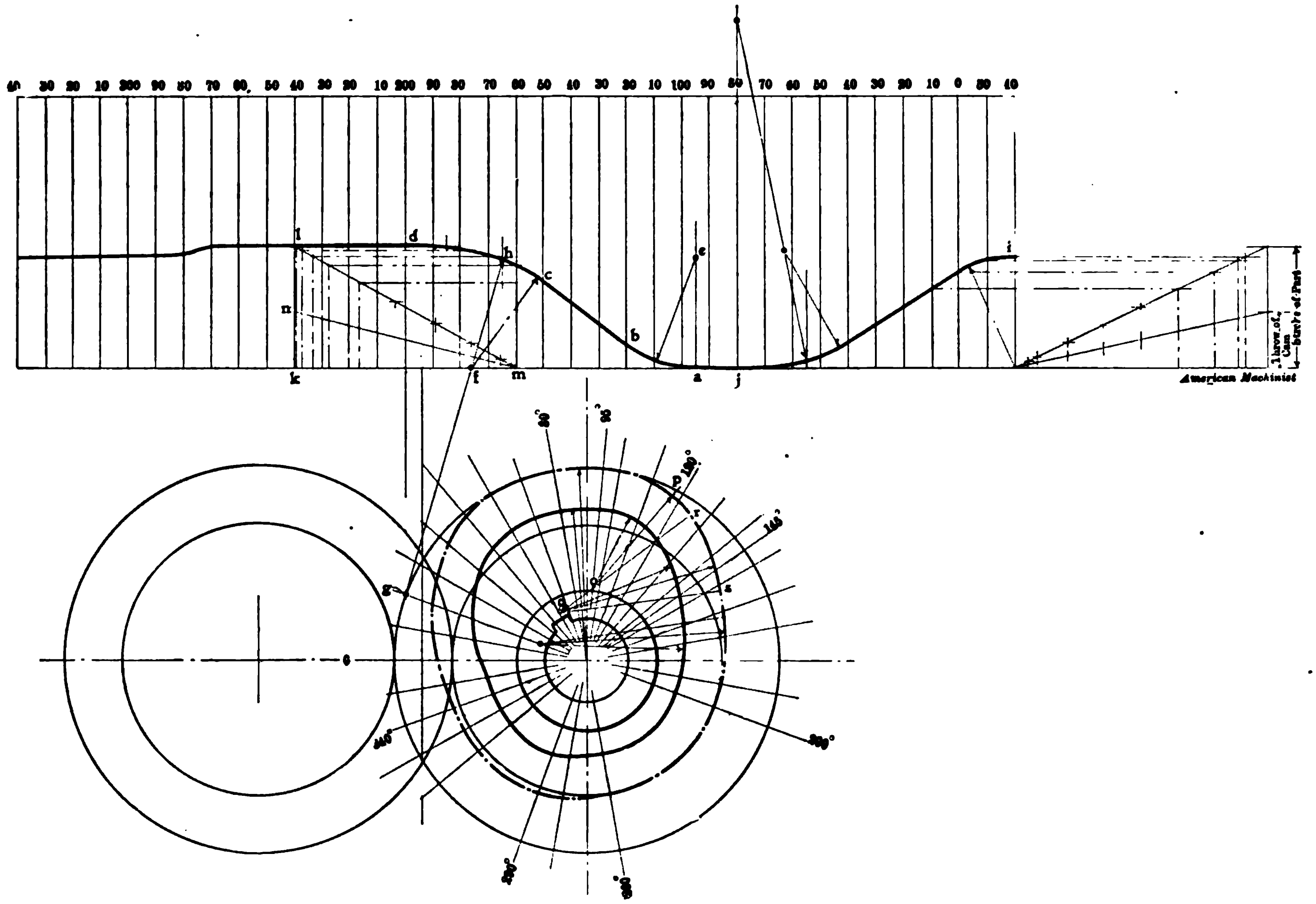


FIG. 50 Laying out the Monotype Cam Curves

determined, the former is then dressed down to it by hand, the greatest and least radii being made to micrometer measurement, since the form of the cam obviously permits the application of the micrometer, the anvil being placed in the hole and the screw being applied to the outer surface. Should the cam make two or more steps in its movement, these steps are invariably accompanied by dwells which involve circular arcs in the cam profile and hence all the steps may be determined with micrometer accuracy.

THE CAM-CUTTING MACHINE

The master former completed, the working formers are made in a cam-cutting machine, made by William Sellers & Co., of which the essential parts are shown in Fig. 52. This machine is regularly used for cutting the cams as manufactured for the Monotype, for which purpose two conjugate formers are used; but it also produces these working formers from a single master former, its working being such as to insure that these working formers are true conjugates.

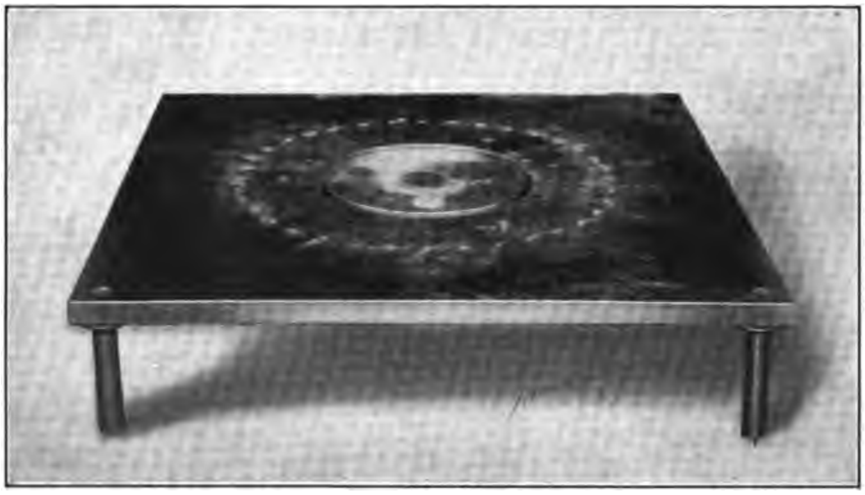
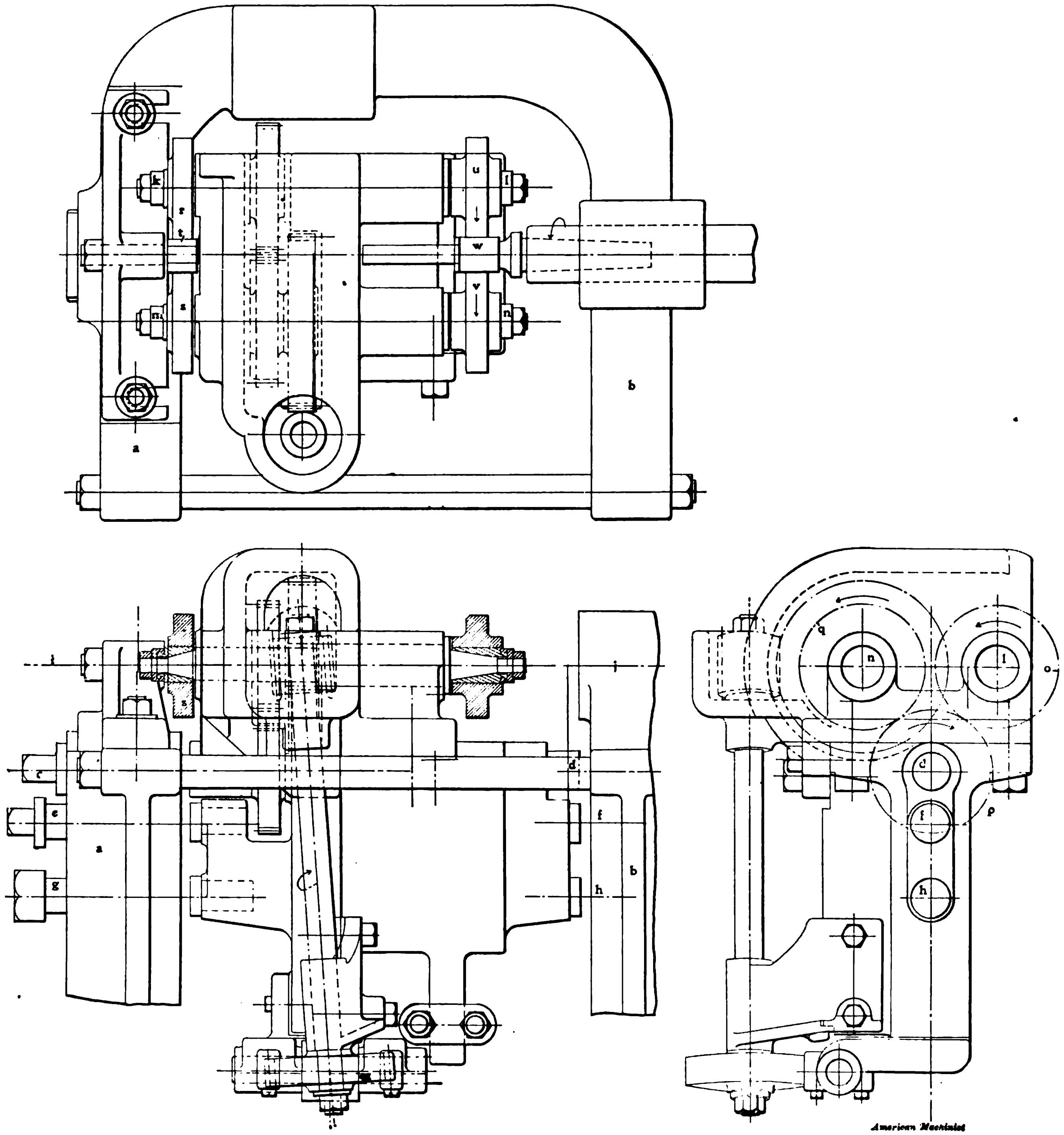


FIG. 51 Iron Drawing Board for Laying Out Master Formers

The main frame of the machine is only indicated at *a b*. Within this frame a rigid box swings on trunnions *c d*, *e f*, or *g h*, the distance of these trunnions from the cutter spindle center line *i j* being equal to the lengths of the cam ends of the cam levers, and since, among the levers, three such lengths are made, three sets of fulcrums are provided. In the illustration the upper set is in use, the others being withdrawn. The box carries two cam shafts *k l* and *m n*, which are slowly revolved by worm gearing indicated by dotted lines, the final connection with the main driving gears being by a flexible shaft to permit the swinging of the box on its fulcrums. The two cam shafts are made to revolve in the same direction and at the same speed by gears *o p q*, and the working formers, which we will suppose have been made, are mounted at *r s*, with a roller pin *t* attached to the frame between them. The cam blanks are at *u v*, and the milling cutter at *w*. As the formers revolve they obviously cause the box, their shafts, themselves, and the blanks to swing on the fulcrums, while the blanks are cut by the cutter, which is of the same diameter as roller *t*, to the same form as the formers. The making of the working formers



American Machinist

FIG. 52 Double Cam Cutting Machine

from the master formers is done in precisely the same manner, except that, there being but one master former, the swinging box is weighted during this operation to cause the master former to maintain contact with the roller pin *t*. A single master former is thus made to produce a pair of working formers, which, from the manner of their formation, must be true conjugates, and, when placed on the former ends of the shaft, must work properly with the roller and without introducing slack in the movement of the swinging box, and thus reproduce their forms indefinitely.

The system of cams described, together with the machine for making them, originated with J. Sellers Bancroft, general manager of the Lanston Monotype Machine Company.

CHAPTER XII

CAM MOVEMENTS OBTAINED FROM BASE CURVES OTHER THAN THE CIRCLE[1]

I was once employed by a printing press manufacturing company who had a lively time with a fly cam. This is the cam which, in conjunction with a spring of some length, operates the fly, delivering the sheet after it is printed. This cam seemed to have the same objectionable feature as with some of the cams in other makes of presses, but somewhat more pronounced. Slap, bang; slap, bang; it seemed as if the fly would pound itself to pieces on the rest rod, and also when delivering the sheet. Something had to be done. Evidently, it was thought, the movement could not have been developed to embody the crank functions of acceleration and retardation, yet on reference to the drawings it was found that it had been, so it seemed that the plan of cut and try with a hardwood pattern until a smooth working form was developed was the only solution, which is a style of cam development frequently in use. It has its demerits as well as its merits, for many valuable movements have developed out of wood whittling.

It occurred to me, nevertheless, that some form other than a semicircle, divided into a number of equal parts and ordinated, could be substituted to give the desired effect. After some consideration the ellipse presented itself to my mind as giving a slower approach to the terminals than that of the crank, although moving quicker in the central position, which in this case did not matter much. A development on these lines was made and gave excellent results. I have used it also on other cam formations and it has been equally satisfactory.

In my many years of practise I have not met with any work on cam development using this elliptic method. Fig. 53 shows both layouts, namely, crank and ellipse, the developments being placed one over the other that the difference may be seen. Line *A* is laid out from the circle and *B* from the ellipse.

CIRCULAR ARCS AND THE ELLIPSE AS BASE CURVES FOR CAM PROFILES[2]

In choosing the character of motion of the roller which is operated or controlled by a cam, it is quite common to choose the crank-motion curve, used by Mr. Smith. The reasons for choosing this motion are that the point moved, starts from rest, is gradually accelerated to the center of throw, then as gradually retarded the same as the crosshead of a steam engine fitted with a Scotch yoke or slotted crosshead. This ordinarily works very nicely, but there are cases where a motion developed from some other curve will give better results.

---

[1] By W. B. Yates.

[2] By Wm. V. Lowe.

In order to make this clearer, I will state a little difficulty I had recently, and how it was overcome. In a certain machine which I designed, it was desired to move a cross-head, or slide, guided in ways, back and forth, and it was necessary that at one end of its traverse it should come very accurately to a definite location every stroke. Some little variation could be allowed on the back stroke, but it was desirable to also keep this nearly uniform. The motion was to start from the back position, move rapidly forward to the front position, dwell for a very small fraction, then return and dwell while other parts of the machine performed their movements.

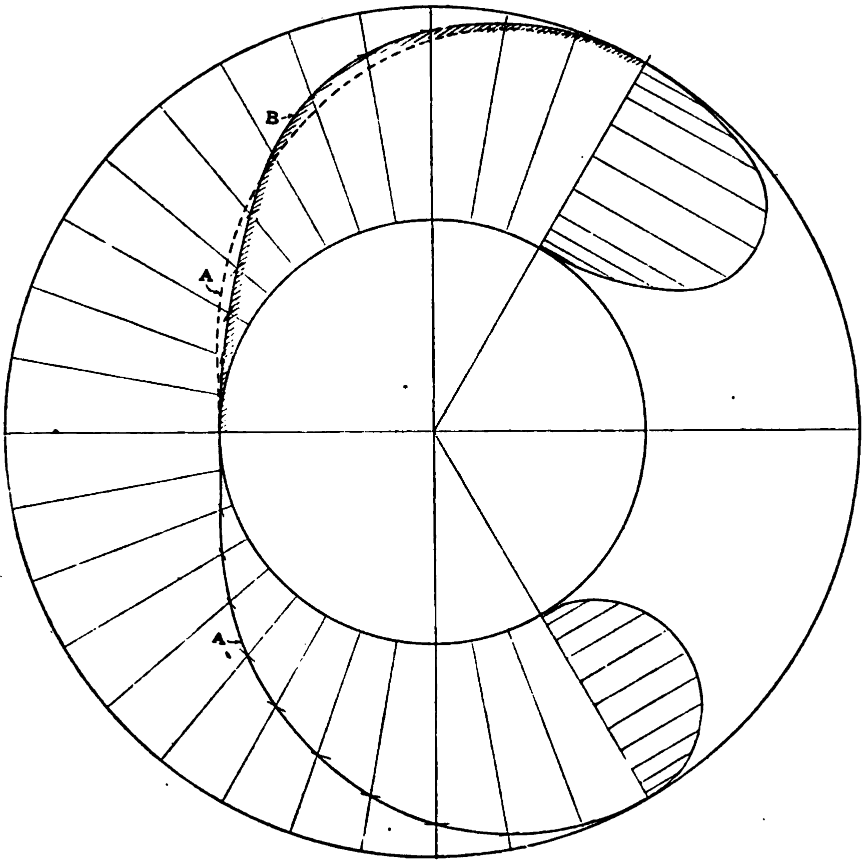


FIG. 53 The Ellipse as a Base Curve

The movement of the slide was 34 inches, and this was derived from a cam of 4 inches throw, the motion of the cam being multiplied by levers to give the 34 inches. The cam was 14 inches diameter, of the face variety, with a roll of 1½ inches diameter. The machine was wanted to make thirty to forty strokes of this slide per minute.

The groove for the roll was cast, and the roll was turned to fit the tight places of the cam, which would be called a pretty fair cast cam; but when the machine was started it pounded so badly, at even twenty to twenty-five strokes, that it was seen something had to be done at once.

The cam could not be made of larger diameter without changing the frame of the machine; therefore the change had to be made in the path of the cam.

The first cam was constructed according to a crank curve, as shown by curve *C* of

Fig. 54. Dividing the semicircle into twenty equal parts, and projecting these to the base, gives the rise for equal arcs of movement of cam, as shown.

After some thought, the compasses were placed on 0 and 20, and arcs struck with radius 0, 20, which intersect, as shown in curve *B*. By dividing *B* into twenty parts, and projecting as before, we get the base line of curve *B*. By comparing, point by point, with *C*, it is seen that the motion is quite different.

It might at first be supposed that by choosing an even longer radius than from 0 to 20 a still easier motion would result. It will, however, be found that the length of the curve from 0 to the apex increases as the radius increases, and that for any radius equal to 0 to 20 or more, the base will be divided exactly the same.

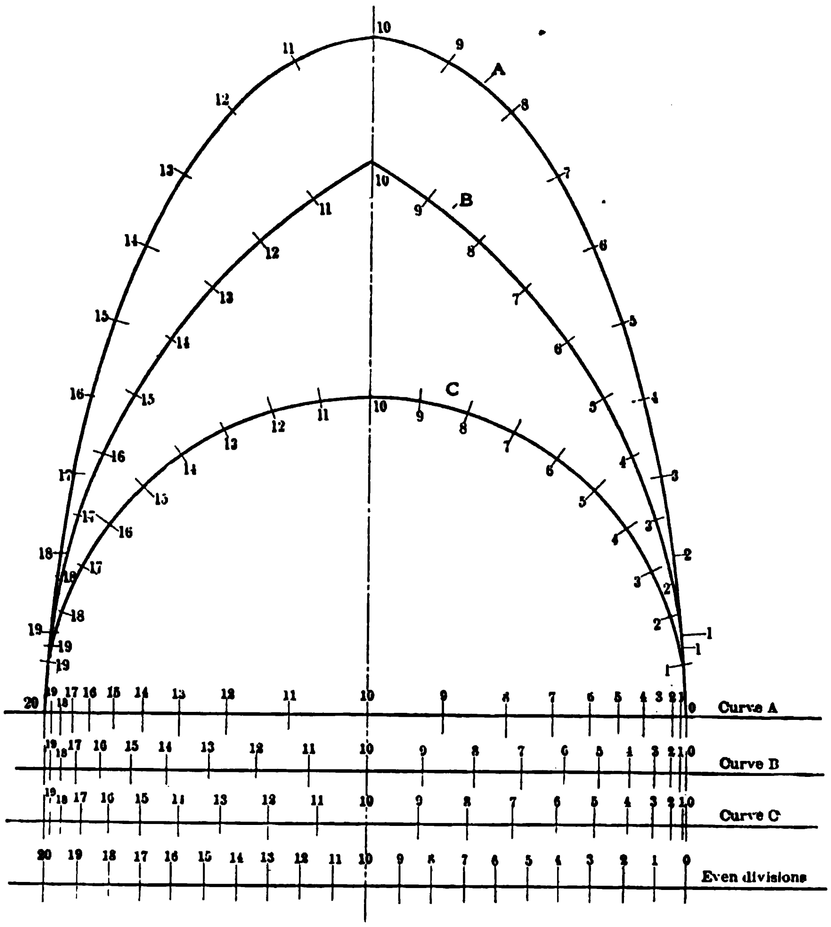


FIG. 54 Various Bases of Cam Curves

A cam was constructed on the base line thus obtained, and the machine ran very much better.

At this time along came a party who owned a similar machine, and claimed he could run it at sixty strokes per minute. Since at this speed the slide would have to make a stroke out and back in about two thirds of a second, and dwell one third, we thought he was exaggerating; further, we had reason to believe the stroke of his machine was less, but we determined to make one more change, since we intended to do *as well* at least as other people.

In the first two cams the roll moved from the outside toward the center of the cam in order to move the slide forward, which gave some lost motion at the extreme forward point. The new cam was modified, so that the roll moved from the center of the cam toward the outside when moving the slide forward, which nearly did away with lost motion at that point, where it was most objectionable. By changing the frame slightly, we made room for a cam 18 inches diameter, and we increased the throw to 5 inches, which cut our ratio of leverage down slightly, and the cam roll was increased in diameter to 2 inches, and then came the question of how the base line or motion line should be divided.

It was thought that if a parabola were chosen the roll would move according to the law of falling bodies, and would be retarded by the same law. On trying this curve, it was thought the change in velocity from zero to the center was too great, and that it was not slow enough at the ends. While thinking this point over, our eyes fell on an irregular curve which looked about right, and curve *A* was drawn in, divided as before, and gave the base line marked curve *A*, and the cam was made to this curve.

The other cams had given so much trouble that we determined to see this one properly fitted up, so we stood by the workman while he filed and scraped this cam till the roll fitted close all the way around.

There is no question but that the action of the machine would have been improved in the first place if the cam had been properly milled in a cam cutter, using a leader; but we did not have either, and it was not the practise at this shop to fit up cams this way.

When this cam was started up the machine ran so smoothly that, with the back turned to it, one could not tell if it was running; consequently, it was speeded up, and by adjusting a little it ran up to sixty strokes per minute, and as smoothly as one could wish. At this point other parts limited the speed and not the action of the slide.

On examining the base line, it is seen the speed is started very slowly, becoming very fast at the center, and is then gradually arrested. With the crank curve the speed was too high at the start and finish, causing the banging at the ends.

The curve finally chosen was nearly an ellipse. The curve *B* has the advantage of being very easily constructed, and is intermediate in rate between *A* and *C*, and would often be good enough and better than the crank curve.

I might add that the performance of the first machine with the last cam was so satisfactory that three machines were added to the original order of three, and this was followed with a repeat order for three more, making nine machines built to date, in addition to the experimental machine.

The inability to run the machine at high speed at first threatened to make it a failure.

Care must be taken that the path from 9 to 11 is not at too great an angle.

THE GRAVITY CURVE AS A BASE FOR CAM PROFILES[1]

The effect of the use of the ellipse instead of the circle as ingeniously used by W. B. Yates is to start the motion of the follower more slowly and to end it with less velocity than with the unsatisfactory harmonic curve, based on the circle, while the velocity at or near the middle of the path of the follower will be considerably higher. In other words, allowance has been made to both start and stop the follower with the least possible velocity consistent with a manageable velocity at the mid-path position.

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[1] By B. A. Lenfest.

Certainly, Mr. Yates has offered valuable testimony as to the satisfaction derived from the ellipse curve method of cam design. There is reason to believe, however, that the ellipse is not the best curve to use in this case, nor can it be defended on scientific grounds — that is, by the application of the laws of mechanics.

There is, however, another curve that can be applied that has a scientific basis, namely, the gravity curve, or the curve of uniform acceleration and retardation. It is desired, herein, to explain this curve as well as the others mentioned in a comparative way. One of the best treatments of this subject is found in Schwamb and Merrill's "Mechanism," recently published by John Wiley and Son. At page 64 is given a defense on scientific grounds for the use of the uniformly accelerated and retarded form of curve. It is believed, however, that a few words more will make the connection clearer, and these words it is the purpose of this discussion to furnish.

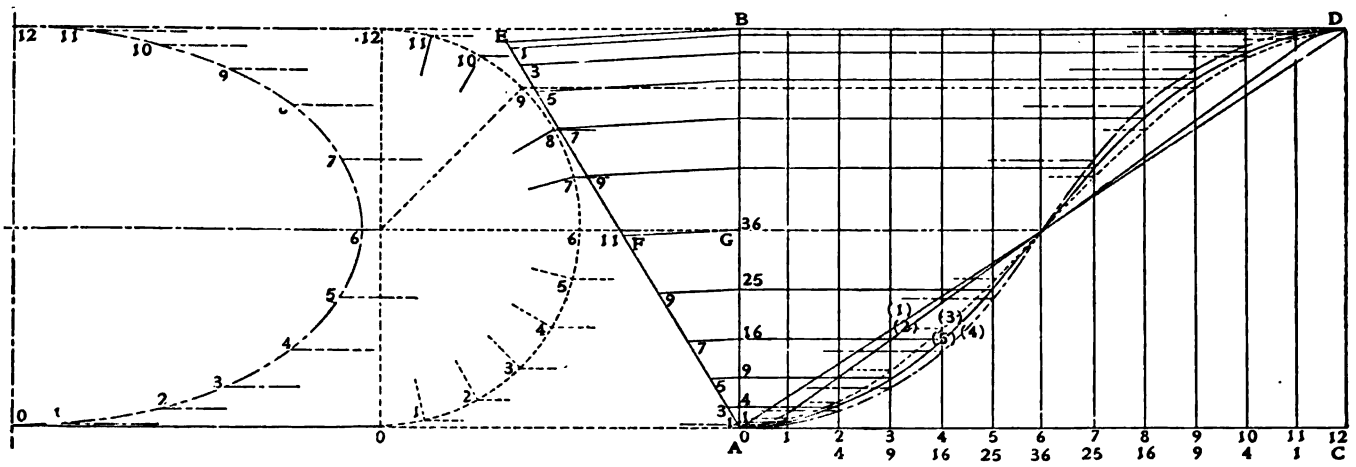

FIG. 55 Various Forms of Cam Curves

VARIOUS CAM CURVES

Referring to Fig. 55, suppose it be desired to move a follower from *A* to *B* at a rapid rate by pushing the cam *A C D* (in this case a translation cam) to the left. There are shown five methods of accomplishing this:

1. The follower may be started at the velocity it is to have throughout the motion, in which case the outline of *A D* will be a straight line (1). This will naturally give a sudden and injurious shock at the start *A*, and there will be a tendency, dependent on the speed, to move above *B*. Both effects are fatal at high speeds.

2. Some improvement can be obtained by using the outline (2) which is a straight line ended by arcs of circles tangent at *A* and *D*, but the curved portion must take care of the entire increase of velocity from 0 to the maximum, a much less efficient method than to distribute this acceleration over the half path of the follower.

3. The dotted curve (3) may be used, which is the harmonic curve that Mr. Yates

has found unsatisfactory.[1] It is constructed, as shown in the figure, thus: Divide the semicircle (dotted) into twelve equal parts numbered 0 to 12; divide the length $A C$ of the cam into the same number of equal parts 0 to 12. Then project the points from the semicircle to vertical lines drawn from the corresponding points on $A C$. A curve drawn through these points will be the harmonic curve and the follower will be moved from A to B in perfect harmonic action for a given uniform velocity of the cam. It will be noted that precisely the same motion is imparted to the piston of a Clayton air compressor or a Jeffers or Hunnewell steam fire engine, for these types of machines have no connecting rod, but replace this connecting rod by a slotted crosshead or "Scotch yoke," so called.

Just because compression is needed, at high speeds, in these engines or in the ordinary steam engine for smooth running, may be a reason why harmonic motion is not desirable for high-speed cams, where no substitute for compression can be offered.

4. The ellipse curve (4) may be used to form the outline $A D$ of the cam. The construction, shown by dash and dot lines in the figure, corresponds in all respects to the layout of the harmonic curve. It should be noted that the distances 0, 1 — 1, 2, etc., along the ellipse outline are equal arcs. The ratio of axes of the ellipse is 1 to 1¾, chosen to correspond with the proportions used by Mr. Yates.

It is clear that as the ratio

$$\frac{\text{major axis}}{\text{minor axis}}$$

approaches 1, the elliptical curve will approach the harmonic curve, and as the major axis is proportionally increased, the follower is started and stopped with less and less acceleration at the expense of a high velocity at the mid-path position. It will be possible to reach a limit beyond which the question of friction at the middle of the path prevents further increase of the follower's velocity.

5. The gravity curve, or, as more often called, the curve of uniform acceleration and retardation (5), is finally considered. The construction is as follows: Divide $A C$ as for the harmonic curve; draw the line $A E$ at any convenient angle with $A B$ and lay off on it, to a convenient scale, distances from A to F and from E to F equal to the odd numbers 1, 3, 5, 7, 9, 11; connect E to B or F to the middle point G of $A B$ and draw from points 1, 3, 5, 7, 9, lines parallel to $E B$ or $F G$, intersecting $A B$ at 1, 4, 9, 16, 25, 36. Project these points thus found on $A B$ to verticals from points 1, 4, 9, 16, 25, 36, on $A C$, and draw the curve (5) through these points thus located on the verticals.

This curve can be defended on scientific grounds as embodying the laws of falling bodies (note the word gravity), such laws being applicable to bodies having a uniform acceleration and retardation due to gravitation. Kent, at page 424 of his Pocket Book, under the title "Falling Bodies," has given the formulas desired.

If g = uniform acceleration due to gravity,
v = velocity at end of t seconds,
h = total height or space passed over in time t,
u = space for each second,
$v = g t$ and $h = \frac{1}{2} g t.$[2]

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[1] And is the one used by Mr. Smith.

Or tabulating:

For t seconds	1	2	3	4	5	6
$g =$	1	1	1	1	1	1
$v =$	1	2	3	4	5	6
$u =$	1	3	5	7	9	11
$h =$	1	4	9	16	25	36

That is, spaces passed over in successive seconds are as the odd numbers 1, 3, 5, 7, 9, 11, etc., while the total spaces passed over from the beginning of motion are as 1, 4, 9; 16, 25, 36, etc.; that is as the squares of the times, as suggested in the formula: $h = \frac{1}{2} g t^2$

A reference to the figure will show that in all respects our cam has been laid out to correspond to the laws of uniform acceleration and retardation, and just as a ball thrown upward will come to rest without shock, so it is reasonable to assume that a cam laid out according to the laws that apply to the ball will cause the follower to start and stop at the ends of the path *A B* with the least possible shock.

CHAPTER XIII

THE LOCATION OF LEVER FULCRUMS FOR FACE CAMS[1]

I AM much surprised at the advocacy of a location for the fulcrum of the lever for a face cam such that the center of the roll would pass through the center of the cam as in Fig. 56. Textile people are frequently compelled to design a cam where such a course would produce working angles that would cause the cam and lever to lock when the cam rotates toward the fulcrum of the lever. Fig. 57 shows the approved textile style in which a straight line passing through the two extreme positions of the cam roller also passes through the center of the cam shaft, and certainly the lines of force are better — *i.e.*, more favorable — with this construction.

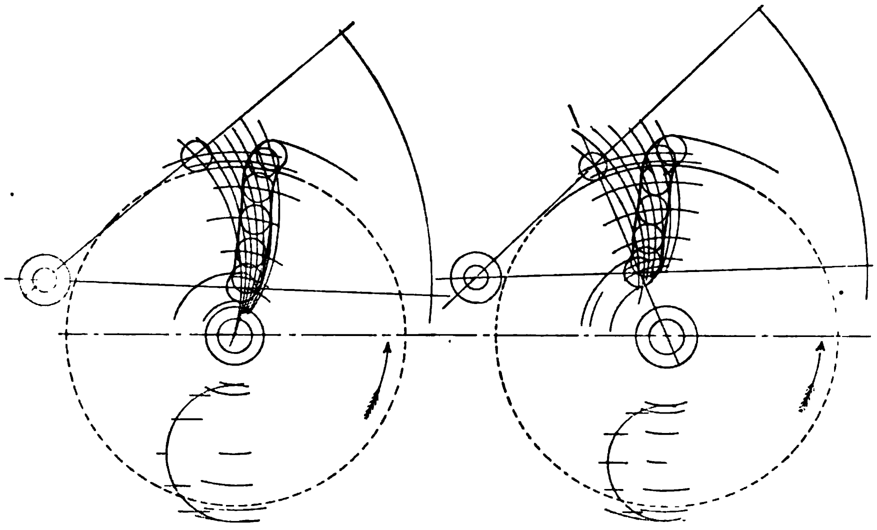

FIG. 56 FIG. 57

The Location of Lever Fulcrums for Face Cams

THE LOCATION OF LEVER FULCRUMS FOR FACE CAMS[2]

Differences of opinion exist anent the corrent location of a cam runner which moves in an arc of a circle and is operated by a face cam; which differences seem to be principally due to the assumption of non-uniform governing factors. Fig. 58 shows arrangements numbered 1 and 2, each of which has supporters, and each of which may be correct, depending on the point of view.

Cam levers of like lengths, having like degrees of angular motion imparted in given timed relation from cams of like diameters, occupying the relative positions 1 and 2, each possess advantages within their respective spheres. The main factor governing their relative efficiency is the duty the lever has to perform, for instance, under two different sets of conditions — first, that of moving a lever against a resistance heavier in the one direction than in the other; second, that of moving a lever to and fro against a constant resistance.

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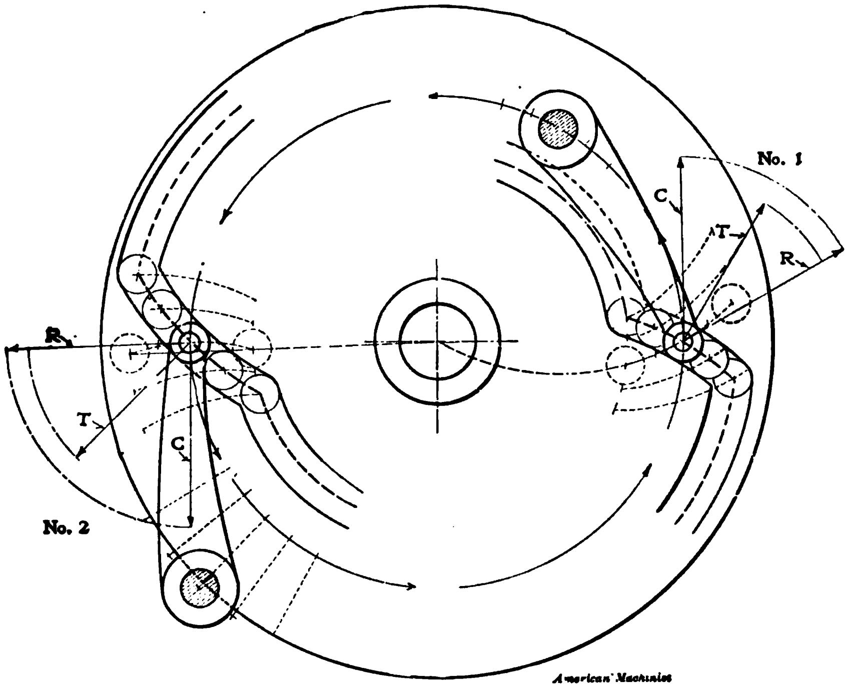
[1] By Clinton Alvord.

[2] By E. A. Howell.

When dealing with the first set of conditions, position No. 1 would be the most favorable, because the mean direction of traverse *C* of the cam more nearly coincides with the direction of traverse *R* of the runner, whereby the length of the acting cam face necessary to give the required motion, together with the relative motion of the frictional surfaces, is reduced; and, further, because the line of thrust *T* of the cam face also more nearly coincides with the direction of traverse *R* of the runner, whereby the working pressures are lessened.



American Machinist

FIG. 58 Location of LevercFul rums for Face Cams

When dealing with a constant resistance position No. 2 is a more correct one, because, with the resistance of the lever to motion equal in either directions, the thrusts and direction of motion of the cam should be equally favorable to traversing the runner either inwardly or outwardly, and this is more nearly the case than it is in position No. 1, in which it is obvious that while the thrust and direction of motion of the cam would be more favorable to the motion of the runner in the outward direction, it would be much more favorable to its return movement. In the case of the resistance to traverse of the runner being in the direction of motion inward, the relative positions of cam and lever would require to be altered to a still further degree from position No. 1, to give the best result.

CHAPTER XIV

MINUTE ADJUSTMENT OF CAM LEVER MOVEMENTS[1]

ALTHOUGH the writers of the preceding chapters have entered very fully into the subject, no instance has been given where wear of the cam roller or its bearing pin would not affect the motion given to the lever or slide, as the case might be. There are cases where such variation is not permissible at all, without entirely spoiling the working of the machine or causing a break. Now it is impossible to prevent wear, and so some other means must be adopted, and a useful purpose may be served by describing one instance on the Linotype composing machine, in which the difficulty has been very simply overcome.

This machine is undoubtedly one of the most ingenious ever made, and the almost human movements of its various parts are produced by levers, in several cases actuated by springs, while the times and lengths of movements are controlled to a great extent by cams; but a special point to be noted is that it is in most cases possible to minutely adjust the relation between the working end (if one may so describe it) and the cam roller, as will be shown in detail later. In some cases it is also possible to adjust the length of stroke of the working end by means of the cam-roller pin, which is eccentric on the portion where the roller runs, as compared with its bearing in the lever. A simple arrangement of such a pin and roller is shown in Fig. 59, which requires no explanation.

The writer was for ten years chief draftsman at the Linotype Company's works in Manchester, and during that time designed a larger size than the ordinary machine, which was first made in the United States. The large machine turns out the "line o' type" for books or magazines any length up to 42 pica ems (7 inches) and any face up to "English" (practically 3-16 inch thick), which is a large piece of metal to produce with perfect type characters all along one edge. However, to return to the question of cams and levers: In one position in the machine the line of matrices (from which the cast is made) is transferred endwise from a box to a bar with teeth, and, later on, from this bar to another one which also has teeth. Unless the parts are in absolute ali[illegible]ment, one with another, the matrices cannot be transferred, and the difficulty is gotten over in the following manner:

Figure 60 shows roughly at *A* the matrix bar in its top position. *B* is part of the frame of the machine and is shaped at *C* for *A* to bed into. *A* is pivoted at *D* to subsidiary lever *E*, which is pivoted at *F* to main lever *G*, of which only the center line is shown, this being sufficient for our purpose. Spring *H* tends to keep *E* down on adjustable stop *K*, which is in the main lever *G;* another spring *L* keeps *A* tilted so as to lead into *C* properly.

Cam 1 actuates lever 2 through roller 3, but this lever is not rigidly fixed to *G*, a spring being interposed in pockets *M* on lever *G* and 4 on the lever 2. A bolt is passed through

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[1] By F. C. Dolby.